USAAVLABS TECHNICAL REPORT 67-40

INVESTIGATION OF HYDRAULIC POWER TRANSMISSION SYSTEMS FOR V/STOL AIRCRAFT

By

J. L. Overfield

H. R. Crawford

August 1967

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DA 44-177-AMC-333(T)

THE WESTERN COMPANY OF NORTH AMERICA
RESEARCH DIVISION

Distribution of this document is unlimited

U. S. ARMY MATERIEL COMMAND

86

BLANK

PAGE

Disclaimer

When Government drawings, specifications, or other data are used for any purpose other than in connection with a definitely related Government procurement operation, the United States Government thereby incurs no responsibility nor any obligation whatsoever; and the fact that the Government may have formulated, furnished, or in any way supplied the said drawings, specifications, or other data is not to be regarded by implication or otherwise as in any manner licensing the holder or any other person or corporation, or conveying any rights or permission, to manufacture, use, or sell any patented invention that may in any way be related thereto.

Disposition Instructions

Destroy this report when no longer needed. Do not return it to originator.

DEPARTMENT OF THE ARMY
U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA 23604

This report has been prepared by the Western Company under the terms of Contract DA 44-177-AMC-333(T). It consists of the results of a feasibility study to determine if a hydraulic transmission system utilizing the latest technology can be effectively employed as the main propulsion power transmission system for Army helicopters.

This command generally concurs in the conclusions made by the contractor. However, considerably more development of pumps and motors must be accomplished before the efficiencies quoted by the contractor can be achieved.

Task 1M125901A01410
Contract DA44-177-AMC-333(T)
USAAVLABS Technical Report 67-40
August 1967

INVESTIGATION OF HYDRAULIC POWER TRANSMISSION SYSTEMS FOR V/STOL AIRCRAFT

Final Report

by

J. L. Overfield
H. R. Crawford

Prepared by

The Western Company of North America
Research Division
Richardson, Texas

for

U. S. ARMY AVIATION MATERIEL LABORATORIES
Fort Eustis, Virginia

SUMMARY

This report covers the results of a feasibility study conducted to determine if a hydraulic transmission system can be successfully employed as the main propulsion power transmission system for Army helicopters. The program approach for the feasibility study is described by the following steps:

1. Search industry and aerospace field for potential hydraulic components which could be used in a transmission system.

2. Of the components reviewed in the search, determine the best components for the system.

3. Determine efficiency, weight and size of the hydraulic transmission system using best components. Make complete layout of system.

4. Determine effect on aircraft components, performance and effectiveness as result of change from a gear to a hydraulic transmission.

The study compares the hydraulic transmission system efficiency, weight and aircraft effectiveness to that of a gear transmission system. The results show the hydraulic system to be competitive with the gear transmission efficiency. The hydraulic system is lighter in weight than the gear transmission and the components replaced by the hydraulic system. An aircraft effectiveness study was conducted to obtain a numerical indication of the improved effectiveness which could be made on a helicopter when using the inherent characteristics of a hydraulic transmission. As an example of what the flexibility of design of a hydraulic transmission can do, on the UH-1F, useful engine horsepower could be increased approximately 3 percent to 5 percent. The infrared signature of the aircraft can be reduced. The foreign object damage (F.O.D.) protection can be increased and engine/airframe vibration reduced.

The hydraulic transmission system described in this study is competitive with a gear transmission system efficiency and weight because it uses three recently developed advances in fluid systems technology. They are:

1. Improved hydraulic fluids. These fluids reduce flow losses by 60 percent to 80 percent in pipes. They improve pump and motor mechanical efficiency by reducing fluid losses in flow passages. The fluids also improve pump and motor volumetric efficiency.

2. A high-efficiency hydraulic pump which operates at jet engine turbine speed (approximately 20,000 rpm).

3. A high-efficiency, lightweight hydraulic motor which operates at aircraft rotor speed. Use of an oscillating cylinder eliminates piston side load losses while multiple rows of cylinders balance most of the bearing load.

The experimental results, test apparatus and test procedure establishing the performance of the improved fluids are shown in the report body and in an appendix. The performance of the pump as calculated by Battelle Memorial Institute and the experimental checks on these performance calculations are cited. The performance of the motor and the basis for the performance results as done by the URS Corporation are shown.

The vulnerability, maintenance, logistics, and operational problems of autorotation provisions, hydraulic line puncture, corrosion, accessory drives, and part-power operation associated with using a hydraulic rather than a gear transmission system are examined. The results of this examination indicate that these problems will not compromise aircraft operation.

FOREWORD

This investigation was performed under the technical supervision of Mr. Meyer B. Salomonsky of the Aircraft Systems and Equipment Division of the U. S. Army Aviation Materiel Laboratories, Fort Eustis, Virginia. The work was conducted to conform to Contract DA44-177-AMC-333(T) entitled "Investigation of Hydraulic Power Transmission Systems for V/STOL Aircraft." Acknowledgement is made to Mr. J. C. Swain of Battelle Memorial Institute, Columbus, Ohio, who graciously supplied a sizing and performance study of a fixed-displacement version of the Battelle "Turbine Speed" pump for this program. Acknowledgement also goes to Mr. Eli Orshansky of URS Corporation, Burlingame, California, who graciously supplied a sizing and performance study for the URS hydraulic motor used in this program.

BLANK PAGE

CONTENTS

LIST OF ILLUSTRATIONS

LIST OF TABLES

LIST OF SYMBOLS

D	tube ID, inches
ΔP	pressure loss, psi/foot
$\Delta P_{C.L.}$	pressure drop in cooling lines, psi
$\Delta P_{H.E.}$	pressure drop in heat exchanger, psi
f	friction factor, dimensionless
F.R.	friction reduction, dimensionless
HP	horsepower
L	tube length, feet
LK	pump or motor leakage, decimal
N_M	hydraulic motor efficiency, percentage
N_p	pump efficiency, percentage
N_{pp}	system efficiency effect of piping losses on pressure drop in system
N_S	system efficiency effect of scavenge pump, percentage
N_{SP}	efficiency of scavenge pump, assumed equal to .90
N_T	efficiency of the transmission system, percentage
P_c	pressure in cooling system, psi
P_R	Prandtl number
$P_{R.L.}$	pressure in pump system line, psi
Q_c	cooling flow, gallons/minute
R_E	Reynolds number, dimensionless
s	specific gravity of fluid, dimensionless
St	Stanton number
V	fluid velocity, feet/second

DISCUSSION

INTRODUCTION

The purpose of this program is to determine the feasibility of using a hydraulic drive system as the main propulsion power transmission system for Army helicopters. The program study model is the UH-1 helicopter. A review of various transmission systems shows that the relatively high efficiency and light weight of gear transmissions have resulted in gear/shaft transmission systems being used on helicopters in spite of their limitations. These limitations include vibration problems, lack of flexibility in component placement, redundant lubrication systems, problems in shifting power from one drive to another, and problems in coupling and decoupling power. However, once the hydraulic transmission system is competitive with the gear/shaft system in efficiency and weight, its numerous advantages make it a superior system.

This report covers an investigation of the combination of certain hydraulic components into a hydraulic transmission system which is shown to be competitive with gear/shaft transmission systems in efficiency and weight. In addition, the system has numerous advantages over a gear/shaft system.

Recent advances in fluid technology and in the technology of hydraulic pumps and motors provided the technical breakthrough which resulted in a hydraulic transmission system that is competitive with a gear/shaft transmission system in efficiency and weight. The fluid technology advances have resulted in fluids with turbulent flow losses reduced by 60 percent to 80 percent. A high-efficiency pump which operates at turbine speed is incorporated in the system. A low-speed, lightweight, high-efficiency motor is used in the design.

Although the UH-1 helicopter was used as the study model for the design, indications are that compound, or multiengine and multirotor (or propeller), V/STOL aircraft will benefit even more from the hydraulic transmission than the simple UH-1 type helicopter.

POTENTIAL ADVANTAGES OF HYDRAULIC TRANSMISSION SYSTEM

Recent advances in fluids technology and in the technology of hydraulic pumps and motors provided the means to make a hydraulic transmission which can be efficiently employed in V/STOL and turboprop aircraft. The resulting transmission can have significant advantages over the existing gear transmission/reduction gear system. Some of these potential advantages are:

1. Elimination of gear vibration and the ability to vibration-isolate the engine provide:

   - Improved airframe and engine life
   - Reduced pilot fatigue

2. Flexibility of design provides:

- Maximum utilization of space
- Weight savings
- Improved aircraft performance (See page 8)
- Increased F.O.D. protection (See page 8)

3. Redundant lubrication systems are eliminated. Present separate aircraft hydraulic pressure supply system is also eliminated.

4. Commonality of components on several aircraft will provide:

- Reduced field parts inventory
- Reduced mechanic training requirements
- Reduced per-unit cost
- Reduced development time and costs per airframe

5. The ability to replace relatively small major components will improve field maintenance and use rate.

6. Simplified power cross-ducting can be provided on multiengine aircraft.

7. Minimum engine synchronization is needed on multiengine aircraft.

8. Variable speed ratio can be provided simply to optimize rotor/engine matching to improve mission range.

9. Simplified autorotation control can be provided on helicopters.

10. If a connecting tube is hit by hostile fire, the fluid circuit is merely closed off. No vibrating shaft remains as on gear/shaft drive.

11. Simple shift of power from rotor to propeller is possible on compound helicopters.

PHYSICAL DESCRIPTION OF HYDRAULIC TRANSMISSION SYSTEM

Figures 1 and 2 show a schematic of a hydraulic transmission system for the T-53 and T-58 engine versions of the Bell UH-1 helicopter. The basic system operates in the following manner:

1. Battelle fixed-displacement, turbine-speed pump is driven by the jet engine to supply high-pressure fluid at the desired flow rate.

2. The pump high-pressure fluid is delivered to the hydraulic motor to drive the motor which drives the aircraft rotor.

3. After work is absorbed from the high-pressure fluid in the motor, the resulting low-pressure fluid is returned to the pump at a pressure level sufficient to prevent cavitation (220 psi).

4. The leakage from the pump and motor passes into the cooling system and through the heat exchanger (if the fluid temperature is high enough to open the thermal bypass valve). The cooling system operates at 15 psi to eliminate the need for a high-pressure cooler (high-pressure coolers are often subject to leaks).

5. The cooling system flow is forced through the cooling system and into the return line to the pump (220-psi line) by the scavenge pump.

COMPARISON OF HYDRAULIC TO GEAR TRANSMISSION SYSTEM

Table I shows a comparison of the overall system losses of the front drive and rear engine drive versions of the hydraulic transmission and the gear/shaft transmission. The results show that the hydraulic system is competitive with the gear/shaft system.

TABLE I
TRANSMISSION SYSTEM POWER LOSSES

Transmission	Hydraulic T-53 Eng.	Hydraulic T-58 Eng.	Gear
	AT DESIGN POINT CONDITIONS, 6600 PSI, 1500 HP, RATED SPEED		
Total losses, HP	75.8	79.2	75
Pump losses, HP	49.5	49.5	-
Motor losses, HP	23.4	23.4	-
Scavenge pump losses, HP	1.1	1.1	-
Piping system losses, HP	1.8	5.2	-
	AT 50% POWER CONDITIONS, 3300 PSI, 750 HP, RATED SPEED		
Total losses, HP	51.2	55.1	Unknown
Pump losses, HP	27.0	27.0	-
Motor losses, HP	22.3	22.3	-

TABLE I - Contd.			
Transmission	Hydraulic T-53 Eng.	Hydraulic T-58 Eng.	Gear
	AT 50% POWER CONDITIONS, 3300 PSI, 750 HP, RATED SPEED		
Scavenge pump losses, HP	1.1	1.1	-
Piping system losses, HP	1.8	5.2	-

Table II shows a comparison of the overall system efficiency of the front drive and rear engine drive versions of the hydraulic transmission and the gear/shaft transmission. These efficiencies are the result of converting the system horsepower losses of Table I into efficiency terms; i.e., horsepower out/horsepower in.

TABLE II SYSTEM MECHANICAL EFFICIENCY COMPARISONS, POWER OUT/POWER IN			
Transmission	Hydraulic T-53 Eng.	Hydraulic T-58 Eng.	Gear
	AT DESIGN POINT CONDITIONS, 6600 PSI, 1500 HP, RATED SPEED		
System efficiency	95.0	94.8	95
Pump efficiency	96.70	96.70	-
Motor efficiency	98.43	98.43	-
Scavenge pump effect	99.94	99.94	-
Piping system effect	99.88	99.66	-
	AT 50% POWER CONDITIONS, 3300 PSI, 750 HP, RATED SPEED		
System efficiency	93.43	93.0	Unknown
Pump efficiency	96.90	96.90	-
Motor efficiency	97.00	97.00	-

TABLE II - Contd.			
Transmission	Hydraulic T-53 Eng.	Hydraulic T-58 Eng.	Gear
AT 50% POWER CONDITIONS, 3300 PSI, 750 HP, RATED SPEED			
Scavenge pump effect	99.88	99.88	-
Piping system effect	99.66	99.12	-

Tables III and IV show the total weight, a breakdown of the weights of each component, and the weight per horsepower transmitted of the hydraulic and gear/shaft transmission systems respectively.

To make a comparison of the weights of the hydraulic transmission to the gear transmission, it is necessary to scale the gear transmission up to the horsepower level of the hydraulic transmission design.

For the front drive system (from Table III),
(.63 lb/hp) (1500 hp) = 950 lb.

For the rear drive system
(.58 lb/hp) (1500 hp) = 855 lb.

The differences between the hydraulic and gear transmission systems at the same (1500) horsepower level are as follows:

Front Drive System		Rear Drive System
Gear System	950 lb.	855 lb.
Hydraulic System	675 lb.	767 lb.
Weight	275 lb.	86 lb.

This greater weight of the gear system can be expressed as an equivalent aircraft efficiency by noting that a percentage point increase in engine efficiency would be equivalent to the aircraft's being able to lift 95 pounds more weight. Dividing the weight increase of the gear system by 95 pounds shows that the hydraulic system improves aircraft efficiency by the following amounts:

$\frac{275}{95}$ = 2.9% efficiency improvement for front drive system

$\frac{83}{95}$ = .93% efficiency improvement for rear drive system

Therefore, it can be said that the hydraulic transmission system, on a weight basis, is lighter or more efficient than the corresponding gear transmission system by the amounts shown above.

TABLE III
HYDRAULIC TRANSMISSION COMPONENTS AND TOTAL WEIGHT

	Front Drive T-53		Rear Drive T-58	
Component	Dia." x L."	Total Weight	Dia." x L."	Total Weight
Pump	8.5 x 12	70.0	8.5 x 12	70.0
Motor	20 x 27	300.0	20 x 27	300.0
Mounting frame	17 x 2	60.0	17 x 2	60.0
Oil cooler	13.2 x 13.2 x 2.5	28.0	13.2 x 13.2 x 2.5	28.0
Piping	-	15.3	-	72.3
Oil	-	25.7	-	69.2
Accessory drive	22.4 x 1	27.1	22.4 x 1	27.1
Scavenge pump drive	-	0.4	-	0.4
Idler gear	-	0.4	-	0.4
Gen. drive	-	0.1	-	0.1
Tach. gen. drive	-	0.2	-	0.2
Valves	-	20.0	-	20.0
Scavenge pump	3 x 4	10.0	3 x 4	10.0
Tail rotor piping	-	30.8	-	24.4
Tail rotor fluid	-	12.4	-	10.0
Tail rotor motor	7.375 x 7.375	75.0	7.375 x 7.375	75.0

TABLE III - Contd.

Component	Front Drive T-53 Dia." x L."	Front Drive T-53 Total Weight	Rear Drive T-58 Dia." x L."	Rear Drive T-58 Total Weight
TOTAL		675.2		767.0
Weight/HP for 1500 HP		.45		.511

TABLE IV
GEAR TRANSMISSION COMPONENTS AND TOTAL WEIGHT

Component	Front Drive (T-53 1100 HP) Weight - Pounds	Rear Drive (T-58 1250 HP) Weight - Pounds
Transmission	425	425
Transmission lube oil	21	21
Transmission oil pump	2	2
Transmission oil cooler	4	4
Speed decreaser	95	105
Shafting, pillow blocks, and supports for speed decreaser to transmission	22	22
Speed decreaser lube oil	7	7
Freewheeling unit	9	9
Hydraulic pump	10	10
Rotor brake	22	22
42 gearbox	21	21
90 gearbox	22	22
Shafting	35	35
TOTAL	695	725
Weight/HP	.63	.58

In order to evaluate numerically the potential advantages of a hydraulic transmission (as listed on page 2), a limited study to determine the expected improvements in the Bell UH-1 helicopter has been made. The results show an example of how the flexibility of design (component placement) of a hydraulic transmission can be used to increase aircraft effectiveness. Other areas of potential improvement also show promise. The results shown are for the Bell UH-1F with the T-58 engine.

1. Eliminating the large, rear-mounted, engine reduction gearbox and side-mounted torque tube and replacing them with a small pump and fluid lines would allow a small, compact, lightweight, bifurcated duct to be used to exhaust the jet thrust of the engine straight to the rear along the centerline of the aircraft. The following improvements would result:

    a. Useful engine horsepower would increase by approximately 3 to 5 percent to:

        Thrust component being along aircraft centerline rather than at angle with centerline.

        Elimination of thrust required to turn present side-directed jet exhaust.

        Elimination of tail rotor horsepower needed to correct side-thrust component.

        Reduction of separation at engine housing boat tail (i.e., reduction of boat tail drag).

    b. The duct would reduce the infrared signature of the aircraft because the duct length would be longer, thus giving more shielding of the hot exhaust gases.

    c. Removing the side running shaft would allow the F.O.D. screen to enclose the engine inlet completely and to eliminate probably the largest source of F.O.D.

    d. Removing the side running shaft would allow the engine to be centered in the inlet plenum to improve inlet recovery.

    e. Eliminating the rear-mounted engine reduction gear would eliminate a redundant lubrication system (the aircraft has a separate lubrication system for the engine, the transmission and the reduction gear).

2. A hydraulic transmission could result in a significant reduction in vibration by allowing the engine to be isolated from the airframe and eliminating the shafts and gears of the transmission. This should result in improved engine and airframe life.

3. Maintenance - Use of separate, smaller transmission components will improve ease of maintenance. A component could be replaced easily rather than repaired on the spot as required by standard gear transmissions.

The major components of the hydraulic transmission have, to some extent, counterparts on a gear transmission system, either by function or location. The comparison is shown below:

Hydraulic system component	Gear system counterpart
Main pump	Reduction gearbox
Rotor motor	Transmission
Connecting piping	Connecting shafts, pillow blocks, freewheeling and articulating drive
Oil cooling system	Oil cooling system
Scavenge pump	Oil cooling and lube pump

The above comparison is included to give the reader more intuitive feel for the operation of the hydraulic transmission system.

VULNERABILITY, MAINTENANCE AND LOGISTICS CONSIDERATIONS

In order to assure that the hydraulic transmission did not cause aircraft safety, vulnerability or logistics problems, numerous items in these categories were reviewed and considerations were included in the design. Some of these are outlined below:

1. Autorotation: To provide rotor autorotation control during an engine flameout or other failure, hydraulic blocking valves located on each side of the rotor motor can be closed by the pilot to cause the motor fluid to recirculate through the motor. An orifice will be used to control the fluid flow rate and thus control the autorotation rate. (See Figures 1 and 2).

2. Hydraulic line puncture: In the event of a hydraulic line puncture, pressure-sensing transducers in the line will close valves to block off the line punctured in order to minimize fluid loss. For critical lines, redundant fluid lines can be used without excessive weight penalty if desirable. When hydraulic lines are struck by hostile fire, a vibrating shaft or gear will not be left in the system, as it is in the shaft/gear system.

3. The hydraulic transmission system is made primarily of aluminum and stainless steel, so there are no new corrosion problems introduced.

4. Provisions have been made to use the existing UH-1 generators and tachometer generator.

5. Provisions have been made for hydraulic power for controls operation during engine failure (autorotation causes the rotor hydraulic motor to act as a hydraulic pump to supply hydraulic pressure).

6. Transmission system losses at partial power conditions (50 percent power at military rated speed) are approximately 0.6 percent less than at 100 percent power. This should compare favorably with the gear system.

DESIGN DATA

The overall efficiency of the system is the result of the combination of the efficiency of the components:

$$N_T = N_p \times N_M \times N_S \times N_{pp} \tag{1}$$

where

N_T = efficiency of the transmission system

N_p = Battelle pump efficiency

N_M = URS Corporation motor

N_S = effect of scavenge pump horsepower on horsepower loss

N_{pp} = effect of piping losses on pressure drop in system

Thus the component losses dictate the system efficiency. Therefore, it was necessary to search for high-efficiency components and to perform optimizing studies to reduce system losses.

A determination of whether to use a hydrostatic, a hydrodynamic or a combination of hydraulic and mechanical drive had to be established early in the design study. A hydrostatic type transmission was chosen after considering the items below.

1. The distance between the driving and driven members was significant in determining that it was more efficient to use a hydrostatic than a hydrodynamic transmission. The piping losses associated with the high-velocity flow of a hydrodynamic transmission in a widely separated driving and driven member would be excessive. Attempting to develop a piping system to diffuse the high-velocity flow to low speed for low losses and then to contract it to high speeds to enter a hydrodynamic turbine would result in high development costs.

2. The relatively large size of a hydrodynamic transmission would be excessively heavy and would interfere with engine inlet performance.

3. Some hydraulic transmission systems use a redundant mechanical drive to improve efficiency at high-speed operation. The high efficiency of the hydraulic transmission system design investigated in this report showed that a redundant mechanical drive was not desirable. Adding a mechanical drive system redundant with the hydraulic system would add excessive weight, would result in the same limitations as for the gear/shaft transmission system, and would not improve efficiency at high-speed operation. A redundant mechanical transmission system normally has advantages only when the engine and final drive operate close to the same speed at high-speed conditions as on a truck or an automobile.

Early in the system design layout stage, certain criteria were established to meet the performance and weight requirements of the system. These are listed below:

1. For a piston-type pump or motor, the cylinder bank must be held stationary to keep windage and friction losses small.

2. Hydraulic fluid characteristics must be improved to keep line, pump and motor weights and sizes small without creating large losses.

3. Fluid passages must be designed as carefully as jet-engine passages to reduce flow losses.

4. The pump and motor must be designed to have the pressure forces in balance to keep main shaft bearing loads and thus bearing sizes at reasonable levels.

5. A pump which operates at turbine speed was desirable to eliminate gears and to keep size and weight small while keeping efficiency high.

6. Hydraulic lines must be optimized to reduce line losses without causing excessive weight penalties.

7. A trade-off study must be performed to determine if system losses are less using a supercharging pump or high pump return line pressure and thus more scavenge pump horsepower drain (to keep the pump from cavitating).

With the above criteria in mind, a search was made to find the best components that industry and the aerospace field had to offer for the hydraulic transmission.

As a result of this search the Battelle Fixed-Displacement, Turbine-Speed Pump was selected for the system pump and the URS Corporation hydraulic motor was selected for the motor of the system.

An explanation of the novel advantages of the pump, motor, and fluids is given below. The experimental and calculated results substantiating the performance of these elements are given in subsequent sections.

The improved hydraulic fluids developed at The Western Company reduce the fluid losses in turbulent fluid flow by 60 percent to 80 percent by laminarizing the flow in flow passages. This is done by building chemical molecular stream tubes in the moving fluid and restricting the random motion of the fluid particles in the direction normal to the fluid flow. Thus the "individual" fluid particles flow in the main direction of flow and do not dissipate energy by particle collision and momentum changes caused by changes of direction of fluid particles in fully turbulent flow.

The Battelle Fixed-Displacement, Turbine-Speed Pump is a rotary-vane pump with novel design features which provide high-efficiency operation at very high (turbine) speeds. A special sizing and performance study on the fixed-displacement pump was performed by Battelle based upon their design and experimental work on a variable-displacement pump (see Appendix I). The significant and novel design features of the pump are described below:

1. The design feature which allows high-speed operation is the pivoting slider foot on the tips of the vanes which rides against the wall of the pump. The foot works on the general principle of the Kingsbury thrust bearing and pivots slightly as load and speed change to provide fully developed hydrodynamic lubrication between the vane tip foot and the wall. Since the foot pivots to provide hydrodynamic lubrication, very high speeds will not generate excessive heat or allow the vane to break through the lubrication film to wear the walls. A laboratory set-up was constructed and tested by Battelle; it established that a vane foot using this principle could support the vane loads and that the coefficient of friction for such a foot would be very low. Thus, heating of the vane tip would not be a problem. (See Reference 6.)

2. The pump has a rotor with a relatively long length compared to its diameter. This allows the pump to pass the large amount of flow required to produce the horsepower specified without having a large diameter which causes higher vane tip speeds. The flow is introduced and exhausted over most of the total length of the rotor.

3. The rest of the pump adheres to the best principles of pump design. The pump flow passages are smooth and well laid out to minimize losses. The pressure drop of the pump was established from experimental model tests. Two pressure-producing pockets or lobes are used so that the pressure forces on the main shaft bearings are balanced. The pump is dynamically balanced.

4. Since the pump operates at turbine speed, it can be very small and still deliver the required horsepower. The hydraulic horsepower equation is a function of flow and pressure. Since the

flow of the pump is a function of rpm and displacement/rpm, having a very high rpm allows the displacement/rpm and thus the size and weight of the pump to be small.

The URS Corporation hydraulic motor is a rotary piston motor consisting of four rows of five pistons each around an eccentric drive shaft. A special sizing and performance study for a 1500-horsepower helicopter motor was conducted based upon the work URS has done on hydraulic motor designs for land vehicles. The significant and novel design features of the motor are described below:

1. The pistons operate in balanced spherical members which eliminate piston side load due to torque reaction. This eliminates the large losses due to piston friction under heavy side loads.

2. The pistons ride on hydrostatic bearings between the pistons and eccentric drive shaft. The coefficient of friction of the hydrostatic bearings is of course very small, resulting in very low losses at these points. The four pads in each hydrostatic bearing also serve to provide the cocking force to rotate the spherical members in which the pistons ride. The cocking force rotates the sphere to aline the piston centerline with the center of the eccentric. This alinement eliminates the side load on the piston.

3. The four rows of five pistons each are placed to balance all but 1.5 percent of the pressure forces on the pistons which would load the eccentric drive shaft. Thus the motor housing bearings are sized only as locating members, not to carry the heavy loads imposed by 1500 horsepower.

4. The motor operates at rotor speed (300 rpm) and is fabricated by relatively lightweight furnace brazing techniques.

DESCRIPTION OF IMPROVED FLUID EXPERIMENTAL APPARATUS

To establish, by experiment, the performance increase which can be obtained with hydraulic fluids in pumps and tubes, a closed hydraulic system (see Figure 5) was set up to measure the improvements. The system consisted of an electric dynamometer driving a positive-displacement pump which forced hydraulic fluid through a measuring section of tubing into an open reservoir. The fluid returned to the pump through a line from the reservoir. For one set of tests, the dynamometer was a constant-output-speed electrical motor. For the variable velocity test, a variable-speed motor was used. The instrumentation consisted of the following items:

1. A torque arm on a dynamometer was used to measure force from which torque and horsepower were calculated. Motor rpm was measured with a strobe tachometer.

2. A positive-displacement nutating disc flowmeter was used to measure flow volume for pump and pipe tests. For the pressure drop test (data in Tables V and VI), the flow rate was determined

by flowing the fluid into a container placed on a double beam balance that was electronically connected to automatically start and stop an electronic timer.

3. The pressure-sensing device used was a Barton differential pressure gauge, which indicated the pressure drop between pressure taps in one reading, thereby compensating for temperature changes and requiring only 0.04 cubic inch of fluid displacement for the total pressure range. The gauge was calibrated with a water and mercury manometer prior to the tests. The pressure taps are 1/16-inch drilled holes which were carefully deburred on the inside of the tubing.

4. Thermocouples were placed in wells in the fluid reservoir and at the discharge from the pump.

For the performance tests, the above apparatus (Figure 6) was used with two different motors. The pressure drop test was performed with a variable-speed motor so that the speed and thus the output of the positive-displacement gear pump was varied. This way it was possible to vary the velocity and the pressure drop in the tube test section. The data on page 48 were obtained from this test setup and are displayed in Figure 7. The other performance test replaced the variable-speed motor with a fixed-speed electrical motor to drive the pump. The data of Figures 8, 9, 10 and 11 were obtained from this test setup.

For the endurance test to determine how the improved fluid would perform for long periods of time in a closed system, the test setup shown in Figure 5 was used. The fluids were pumped with a gear-type positive-displacement pump to a four-inch manifold and on to the test section. The tap was a 1/16-inch hole. One inch downstream of the pressure tap, a hole was drilled to accommodate a small thermocouple which probed about one inch into the pipe. The second pressure tap was 31 feet from the first with another thermocouple placed one inch downstream. Three feet past the second tap the test section increased to four inches for the return to the positive-displacement nutating disc, flowmeter, and inventory vat. This particular system was used for the flow tests.

IMPROVED FLUID EXPERIMENTAL PROCEDURE

For the pressure drop test where pressure drop was measured at various pipe flow velocities, the following test procedure was used:

1. Zero readings for all instruments were made before the variable-displacement electrical motor was started.

2. MIL-H-5606A oil was forced through the test system (Figure 5) with a gear pump driven by a variable-speed electric motor.

3. With the system at steady-state conditions, the pressure drop in the .416-inch ID tubing test section, the reservoir fluid temperature, the time, and the amount and time for this flow were

recorded. These data were reduced to pressure drop and velocity for the curve of Figure 7.

4. The speed of the variable-speed motor was increased to increase the velocity and the pressure drop, and another set of readings was taken. This procedure was repeated until the base fluid curve of Figure 7 was obtained.

5. The system was allowed to cool to the original fluid temperature of step 1.

6. The additives G-5 and G-15 (now designated G-8) were added to the hydraulic fluid, and steps 1 through 4 were repeated.

For the pump and tubing performance test of Figures 8, 9, 10 and 11, a fixed-speed electric dynamometer motor replaced the variable-displacement motor so that motor horsepower could be measured by the torque arm on the dynamometer (Figure 5). The test program followed the steps below:

1. A zero reading was taken before the electrical dynamometer was started with MIL-H-5606A fluid in the system.

2. The electrical motor was started, and when steady-state conditions were reached, a reading was taken.

3. Because the closed system absorbed the energy of the pump, the temperature of the fluid in the system increased. Readings were taken at approximately each 5°F increase in fluid reservoir temperature.

4. The electric dynamometer was stopped and the system was allowed to cool to the original starting temperature. The improved fluid additives were then added to the fluid and the test sequence of steps 1 through 3 was repeated.

For the endurance testing, the improved fluid was circulated for over 200 hours in the test apparatus of Figure 6. The pressure drop in the measuring section was recorded at the beginning of the test and at intervals thereafter. The testing was normally conducted in periods of six to eight hours during the working day.

IMPROVED FLUID EXPERIMENTAL RESULTS

The raw data from the improved fluid tests described above are listed in Appendix I. The parameters reduced from the data of the pressure drop and pipe velocity tests were pressure drop per 100 feet of pipe and fluid pipe velocity in feet per second. These were obtained by dividing the measured pressure drop by the pipe test length in feet, then multiplying by 100 feet. The tube fluid velocity was obtained by measuring the time required for a measured amount of fluid to flow through the system; converting this to cubic feet per second and dividing by the cross-sectional area of the tube allowed calculation of the flow velocity in feet per second.

For the pump and tube performance test, the following relations were used:

$$\text{Volumetric efficiency} = \frac{\text{measured flow}}{\text{ideal flow}}$$

where measured flow was obtained by measuring the time for a measured amount of fluid to flow through the system, then converting to gallons per minute; ideal flow was obtained by multiplying the displacement per revolution by measured rpm.

The pressure drop per length was obtained as in the pressure drop test.

Horsepower was obtained by multiplying the force on the torque arm of the motor by the torque arm length to get torque and then multiplying by rpm to get horsepower (all in appropriate units).

BATTELLE FIXED-DISPLACEMENT, TURBINE-SPEED HYDROSTATIC PUMP PERFORMANCE

In Appendix II, the design point performance for the Battelle Fixed-Displacement, Turbine-Speed Pump is shown for standard fluids and for improved fluids. The improved fluids reduce the 2.0-percent fluid friction loss of the pump by approximately 60 percent to 0.8 percent, and the 2.0-percent volumetric efficiency loss will remain constant. It is assumed that the 0.5-percent loss due to viscous shear is not improved. The resulting overall system efficiency is 97.7 percent.

Performance of the fixed-displacement pump was calculated by Battelle Memorial Institute based upon the results of their tests and calculations done for their variable-displacement pump reported in Reference 6. Figure 12 shows the pump performance as system pressure is changed for the variable-displacement pump (data from Reference 6 and replotted versus pressure). The lower line of the bands of performance represents the experimental or calculated results. The upper line represents the maximum reasonable expected losses of the actual pump. For the fixed-displacement pump at 50-percent power, the trend of performance would be the same as that of the variable-displacement pump, except that pump flow would remain constant as it would for the variable-displacement pump. Thus, flow losses and drag losses would remain constant, but leakage loss would go down as pressure goes lower. Thus, as horsepower reduces as rotor horsepower demands reduce (and thus reduce pump back pressure and thus pump operating pressure), the 2.0-percent leakage loss of the fixed-displacement pump would reduce the same percentage as the variable-displacement pump for the same pressure change. Thus, overall efficiency of the pump would increase by the amount that the leakage losses are reduced. The curve shows that the leakage losses reduce from 2.0 percent to 0.5 percent, going from 6600 psi to 3300 psi, which is a 1.5-percent reduction in losses. This would mean that on the pump, a

$$1 - \left(\frac{0.5}{2.0}\right)(2) = 1.5\%$$

reduction in losses; or overall losses would be down by 1.5 percent at 50-

percent power for the hydraulic transmission due only to the increase in pump performance.

URS CORPORATION HYDRAULIC MOTOR PERFORMANCE

The design point performance, the performance calculation relations, the losses of each point where motor losses occur, and the performance at 50-percent power are shown in Appendix III.

The design point overall efficiency is shown as 98.43 percent. This was obtained by taking the efficiency for the motor, calculated by URS in Reference 10, for standard fluids and then correcting the efficiency for improved fluid effects. The losses which were reduced by the improved fluid were the passage flow losses (HP_{FP}) which would normally have constituted the largest single loss element in the motor. Being a low-speed motor, the losses are already small (low piston speed, low fluid velocity, low valve speed, etc.).

The URS Corporation efficiency results were calculated on the motor performance computer program. The loss relationships are standard hydraulic pump and motor relationships taken from the literature referenced in Appendix III.

The improvements in performance due to the improved fluid were obtained by calculating the Reynolds number in the flow passages and using the fluid friction loss reduction of Figure 13.

CONNECTING PIPES EFFECT ON EFFICIENCY

The connecting pipes which carry the high-pressure fluid from the main pump to the rotor motor and the low-pressure fluid back to the pump, plus the cooling system and leakage pipes, have pressure losses caused by the flow of fluid through the pipes. These pressure losses are greatly reduced by the improved fluids. The total pressure losses in the system divided by the operating pressure constitute the fraction of the energy loss of the system attributed to the piping system.

The pressure losses of each pipe in the system are shown on the tables in Figures 3 and 4 (the drafting layout of the transmission system). The calculation of tubing losses is described below.

Using tube layout of the prototype transmission, a calculation of line and fluid weights for various sizes and types of pipes was made. The pressure losses, and therefore, the system efficiency losses for various size tubes, were also calculated. The optimum between large tube sizes for small efficiency loss and small tube sizes for low weight was established using the sensitivity (weight lifted/percentage point of efficiency) of the UH-1 helicopter.

In order to minimize size and weight of the system, several component locations and tubing layouts were made.

After various tradeoffs of weight and efficiency were investigated, the optimum size of the tubing was established at 2.25 inches ID for the main connecting tubes. The high-pressure line between the pump and the motor, operating at 6600 psi, would be AM 350 (AMC Spec. 5584) seamless stainless steel tubing with ultimate strength of 165,000 psi (double-aged condition) and yield strength of 135,000 psi. With a wall thickness of .188, the tubing would be stressed to 37,800 psi with a resulting factor of safety of approximately 3.5. The line and fluid total weights and the line pressure drops are shown on Figures 3 and 4.

The return, low-pressure lines and secondary (cooling and scavenge system) lines are of aluminum tubing with strength exceeding the stresses imposed by the internal pressure (with a safety factor of about 4).

The line losses were obtained by calculating the pressure losses for the base fluid in lines and fittings from a standard pressure loss equation from Reference 1; namely,

$$\Delta P = .0808 f \frac{L}{D} V^2 s \qquad (2)$$

where

f = friction factor

L = tube length, feet

D = tube ID, inches

V = fluid velocity, feet/second

s = specific gravity of fluid

For fittings and bends, an equivalent length of tubing as described on page 126 of Reference 1 was used.

To determine the pressure loss in the pipes with the improved fluid, the ratio of base fluid losses to fluid losses with additives at a calculated Reynolds number was used, as shown in Figure 13. Not all the lines always have fluid flowing in them. Only lines normally flowing were included in the tabulation.

Loss calculations were done for both the T-53 and T-58 versions of the UH-1.

Equation (2) describes the friction reduction with the improved fluid additives. This equation allows calculation of the friction reductions, which can be obtained over the entire range of flow rates that will be experienced in the hydromechanical transmission. The equation is limited to Reynolds numbers greater than 2000 (minimum turbulent flow Reynolds numbers), since no reduction in friction is expected at Reynolds numbers where the flow is not turbulent. The new equation also fits the mathematical model of the fluid additives. The equation for friction reduction for smooth pipes is:

$$F.R. = (0.8)\,\frac{(.0014 + .125\,R_E^{-0.32}) - 16/R_E}{(.0014 + .125\,R_E^{-0.32})} \qquad (3)$$

where

F.R. = friction reduction

R_E = Reynolds number

ENVIRONMENTAL TEMPERATURE EFFECTS AND HEAT EXCHANGER SIZING

In order to determine the horsepower requirements of the scavenge pump, it is necessary to determine, among other things, the heat exchanger size and thus the heat exchanger pressure drop. To do this, the effects of environmental temperature and heat dissipation from the transmission were determined.

Environmental temperature effects on the fluid temperature and hence on performance and heat dissipation will be limited (for steady-state operation) due to the use of a thermal bypass valve in the oil cooling system. Steady-state operation will be performed predominantly at a constant transmission oil temperature. The prime effect of temperature on the system will be in the sizing of the oil cooler and thus the oil cooler pressure drop and weight. As calculated in Appendix II, the weight of the oil heat exchanger is 27.4 pounds and the pressure drop is 3.9 psi. The flow is eight gallons/minute.

Other effects of temperature considered were:

1. Hot-day (103° F) operation of the jet engine reduces the power output of the engine and thus the heat losses due to inefficiencies which are a percentage of the horsepower output. This tends to compensate for the reduced air temperature gradient available to conduct the heat from the oil cooler. The cooler was sized for 125° F air temperature, as is the present gear system.

2. Beneficial oil cooling will be obtained from the pipes and tubing carrying the fluid. This beneficial effect will not be included in the heat exchanger sizing and will provide a margin of cooling capability.

3. The T-53 reduction gear circulates its oil to the engine oil cooler. Theoretically, the engine oil cooler could be reduced in size, since it will need to cool only the engine oil if a hydrostatic transmission is used. This reduction will not be included in this study. This provides additional margin for the system. In fact, there is the possibility that the existing engine oil cooler may be usable to cool the hydraulic transmission system without the addition of a transmission oil cooler.

4. Oil cooler data obtained from Bell UH-1 project personnel are shown in Appendix IV and were used for basic cooling parameter data.

5. Environmental temperature requirements for the T-53 engine will be limited by the maximum temperature of the incoming air. MIL-STD-210A hot-day maximum is 103° F. The oil temperature may become slightly warmer than 103° F when the aircraft is standing in the direct sunlight; however, when the engine starts, air will flow over the components and their environment will then be at ambient air temperatures. The maximum ambient operating temperature requirement for components on the T-58 engine as listed in the Installation Manual is 200° F. Thus, heat will flow from the transmission to the surrounding air at all times, since the hydraulic transmission operates at 230° F.

The oil cooling system for the hydraulic system consists of the following:

1. Oil cooler similar to the oil cooler existing on the UH-1.

2. Thermal bypass valve to stop heat exchanger oil flow circulation when oil is equal to or less than 230° F.

3. Scavenge pump to gather and force oil through the cooling system.

4. Piping of cooling system.

Effect of temperature on performance of transmission fluid lines is described below.

Figures 14 and 15 show the effect of different transmission oil operating temperatures on the fluid line losses. At the operating temperature, 230° F, there is a .073 percent loss for the T-53 transmission system. During start-up on 0° F days, the line pressure loss will be 0.16 percent until the transmission oil begins to warm to operating temperature. As it warms, the pressure loss reduces as per Figure 14. The losses for the T-58 transmission system line losses are presented in Figure 15.

The environmental temperature effect on the pump and motor performance will be similar to the effect on the system piping. That is, during steady-state operation there will be no change in system temperature due to the thermal bypass valve's holding the system at a constant temperature. During the warm-up period, the pump and motor will be slightly less efficient than at steady state. The warm-up transient temperature does not have a significant effect on the transmission operation.

SCAVENGE PUMP EFFECT ON EFFICIENCY

The scavenge pump must take the leakage flow of the pump and motor, force it through the pipes and cooler of the cooling system, and then pump it into the pump return line (against the pump return line pressure, 220 psi). The

work of the scavenge pump must be charged against the system efficiency. The heat exchanger sizing and flow rate, the pump return line pressure required to eliminate pump cavitation, and the piping in the secondary system determine the scavenge pump horsepower requirements.

The sizing study of the heat exchanger has been used to establish the fact that a supercharger pump is not necessary in the system. A supercharger pump could be put in the system because it is possible that the cooling flow required multiplied by the pump return oil pressure (needed to keep the pump from cavitating) would result in large scavenge pump horsepower levels. Since the scavenge pump merely raises the leakage and cooling flow oil to pump return line pressure from the low pressure of the oil cooler (low pressure to reduce leaking tendency), the scavenge pump horsepower is not useful horsepower to the system. The supercharger pump being in the pump return line system does useful work and only its inefficiency would be charged to the inefficiency of the system.

However, using the heat exchanger sizing, the cavitating pressures and the leakage flow of the pump and motor show that the scavenge pump horsepower requirements will be small:

Cavitation elimination pressure	= 220 psi
- Cooling line pressure level	= 15 psi
Resulting ΔP	= 205 psi
Heat exchange ΔP	= 3.9 psi
Cooling line loss ΔP	= 3.3 psi
Total ΔP across cooling	= 212.2 psi

Then the horsepower loss due to the scavenge pump is:

$$HP = (P_{R.L.} - P_c + \Delta P_{C.L.} + \Delta P_{H.E.})\, Q_c K/\eta_{S.P.} \tag{4}$$

where

HP	= horsepower
$P_{R.L.}$	= pressure in pump return line, psi
P_C	= pressure in cooling system, psi
$\Delta P_{C.L.}$	= pressure drop in cooling lines, psi
$\Delta P_{H.E.}$	= pressure drop in heat exchanger, psi
Q_C	= cooling flow, gallons/minute
K	= conversion factor, .000583
$\eta_{S.P.}$	= efficiency of scavenge pump = 90%

$HP = (212.2 \text{ psi}) (8 \text{ gallons/minute}) (.000583/.90)$

$\underline{HP = 1.1}$

The percent efficiency decrement associated with one HP = $\underline{\underline{+073}}$.

A supercharger pump would cost the system approximately 0.1 percent to 0.2 percent in efficiency.

HYDRAULIC MOTOR MOUNTING FRAME

In order to mount the hydraulic motor which drives the helicopter rotor, a frame was designed that is an integral part of the hydraulic motor housing and mounts on the helicopter frame lugs provided for the present gear transmission.

The weight, size, and stresses in the hydraulic motor mounting frame have been calculated. The hydraulic motor mounting frame is made of C 35-T-61 aluminum stressed to a maximum of 14,800 psi with an ultimate strength safety factor of 2.75. The mounting frame is 25 inches by 32 inches by 3-1/2 inches thick. It mates with the existing airframe mounts. It weighs 60 pounds.

FUTURE HYDRAULIC TRANSMISSION TRENDS

The curves of Figure 19 show how the weight of a hydraulic transmission system becomes increasingly smaller as the system design point pressure is increased. Figure 20 shows how the size of the pump, for example, becomes increasingly smaller as the pump design point pressure is increased. These trends point the way to the future. Increasingly higher operating pressure with smaller weight and size is the direction of future hydraulic transmissions. This will allow greater payloads for V/STOL aircraft. It will mean fewer engine performance penalties on front-drive engines because the inlet airflow will be less distorted with a small pump than with a bigger pump or big reduction gearbox. The blockage area in front of the engine is smaller with resulting performance improvements. The foreign object damage filter area could probably be increased, since the small pump will not interfere with the filter area as much as big reduction gearboxes or larger pumps.

The curve of Figure 21 shows that the system efficiency is not greatly affected as design point pressure changes. This provides the possibility of going to higher operating pressures without excessive penalties in efficiency. In this feasibility study, for example, the overall aircraft performance with the hydraulic transmission could be increased slightly by operating the system at 8000 psi rather than 6600 psi. This increase was not used because of the expected reduction in reliability of associated hydraulic parts (fittings, valves, joints, etc.); but with design improvements in the associated hydraulic components, increased pressure operation has numerous advantages.

The study identifies in a very decided manner the need for the pump of the hydraulic transmission to operate at gas turbine output speed. The resulting light weight, small size, and elimination of a reduction gear with its housing make this requirement a must.

The higher pressures of the hydraulic transmission systems of the future indicate an increasing use of the synthetic hydraulic fluids because of their reduced compressibility and therefore improved hydraulic component performance.

The improved fluids used in this study foreshadow other improvements in hydraulic fluids to improve performance. The improved fluids used in this study greatly reduced flow losses. The synthetic fluids mentioned above would reduce compressibility losses. Another significant step forward would be the reduction of fluid shear losses (without making the fluid so thin as to greatly increase leakage losses). Until now, the losses in a hydraulic transmission have been large compared to some of the losses, like compressibility; but as the magnitude of the system losses becomes smaller, each small improvement becomes increasingly more significant.

The detailed design improvements that develop as a type of device comes into greater usage will continually improve the hydraulic transmission by reducing losses in each of the areas where losses occur by a small amount.

Since operational use of the type of transmission investigated in this study is approximately three years away if the effort is continued, the hydraulic transmission of the 1970-1975 period would be an operational version of this transmission.

In summary, the hydraulic transmission for V/STOL aircraft (using high-speed jet engines as prime movers) would have the following characteristics:

1. Turbine speed hydraulic pump.

2. Lightweight, rotor or propeller speed hydraulic motor.

3. Improved hydraulic fluids with reduced flow losses.

4. Components designed to operate reliably in the 5000-to-7000-psi pressure range.

Future improvements beyond the above would include the following:

1. Higher pressure operation (8000 and higher) to reduce further the size and weight of components.

2. Fittings, connections and accessories such as pumps and motors which can be depended upon to operate reliably at high pressures without excessive leakage problems. Heat exchangers which can operate at return line pressures without leaks (200-psi region).

3. Further advances in fluid technology to improve performance.

4. Fluids with reduced compressibility at high pressures.

SYSTEM RELIABILITY

The system design criterion used for reliability was to design all members to operate a minimum of 1000 hours at the power level and speed described for gear transmission endurance tests. The endurance times at given power levels used are described below:

1. 10 percent of time at 25 percent over rated power at overspeed conditions.

2. 40 percent of time at rated speed at rated horsepower.

3. 50 percent of time at 40 percent to 90 percent of rates horsepower.

TAIL ROTOR DRIVE

To drive the tail rotor, a high-pressure line off the main pump delivers high-pressure flow to a hydraulic motor which drives the tail rotor. A return low-pressure (220 psi) line returns the fluid from the motor to the low-pressure side of the pump. The hydraulic motor used is a Dynex MF-3021 type motor. The layout of the tail rotor system is shown on Figures 17 and 18. The weights and pressure drops in the system are also shown on these figures.

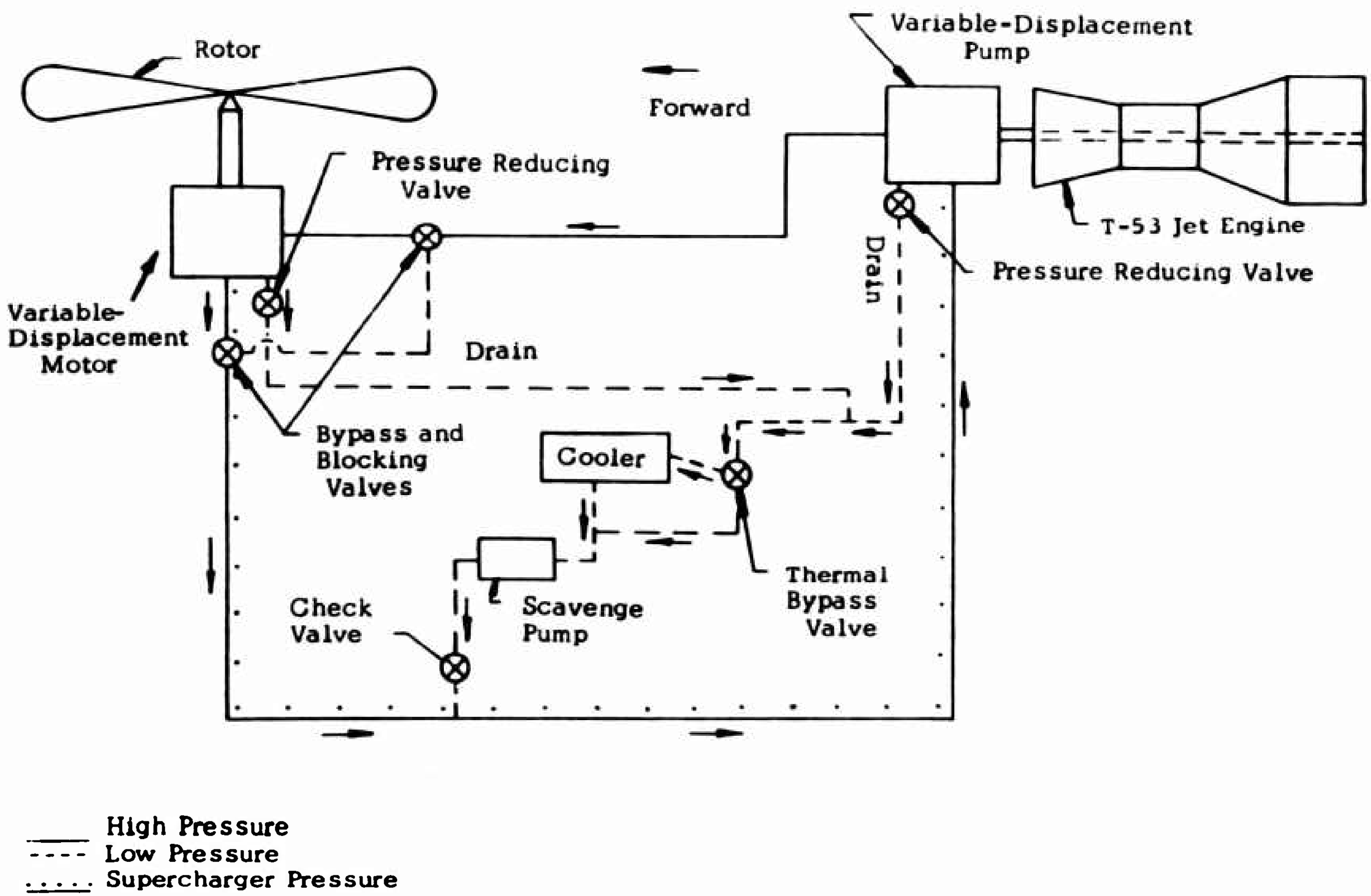

Figure 1. Schematic of Bell UH-1/T-53 Hydraulic Transmission System.

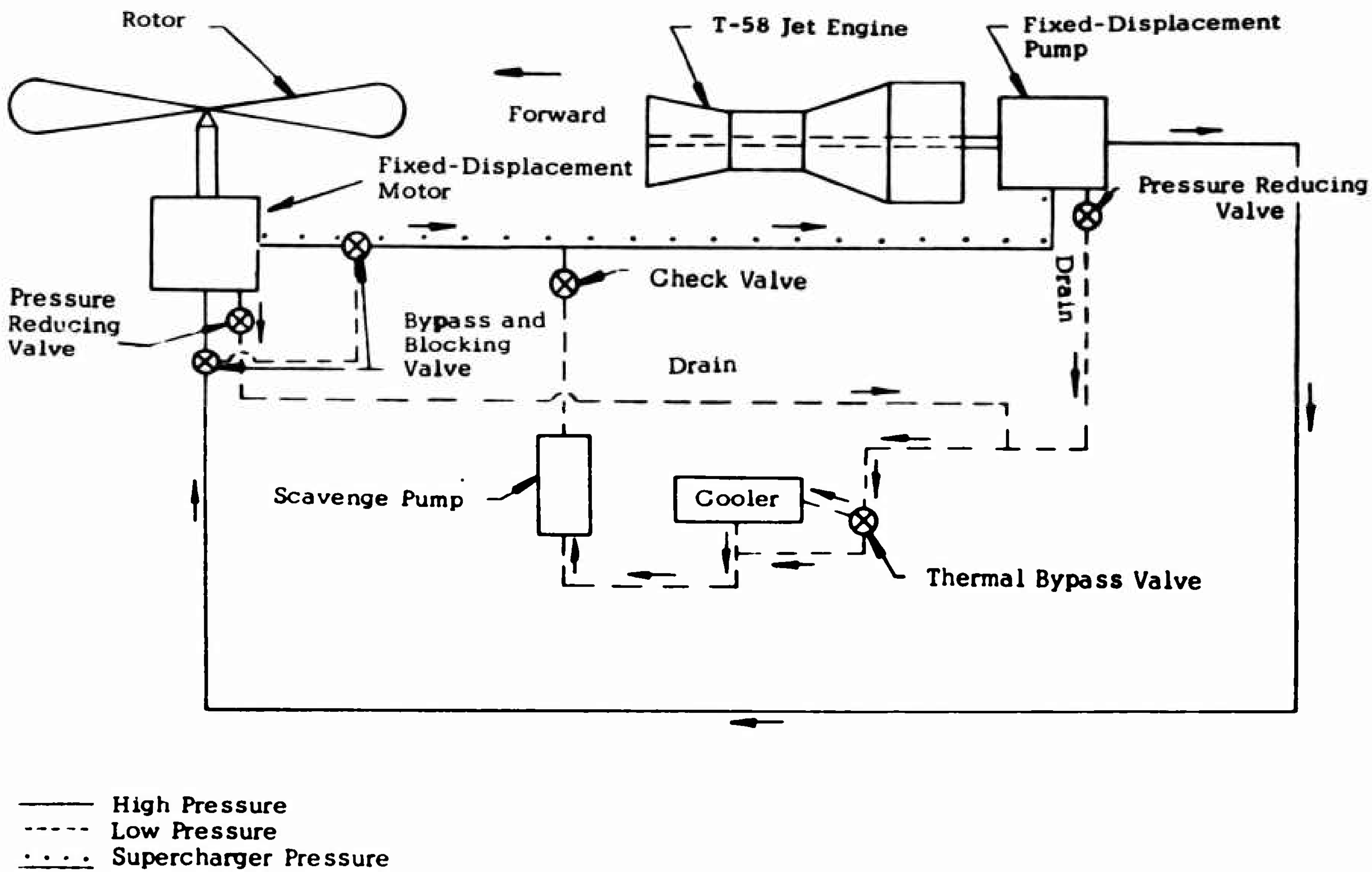

Figure 2. Schematic of Bell UH-1/T-58 Hydraulic Transmission System.

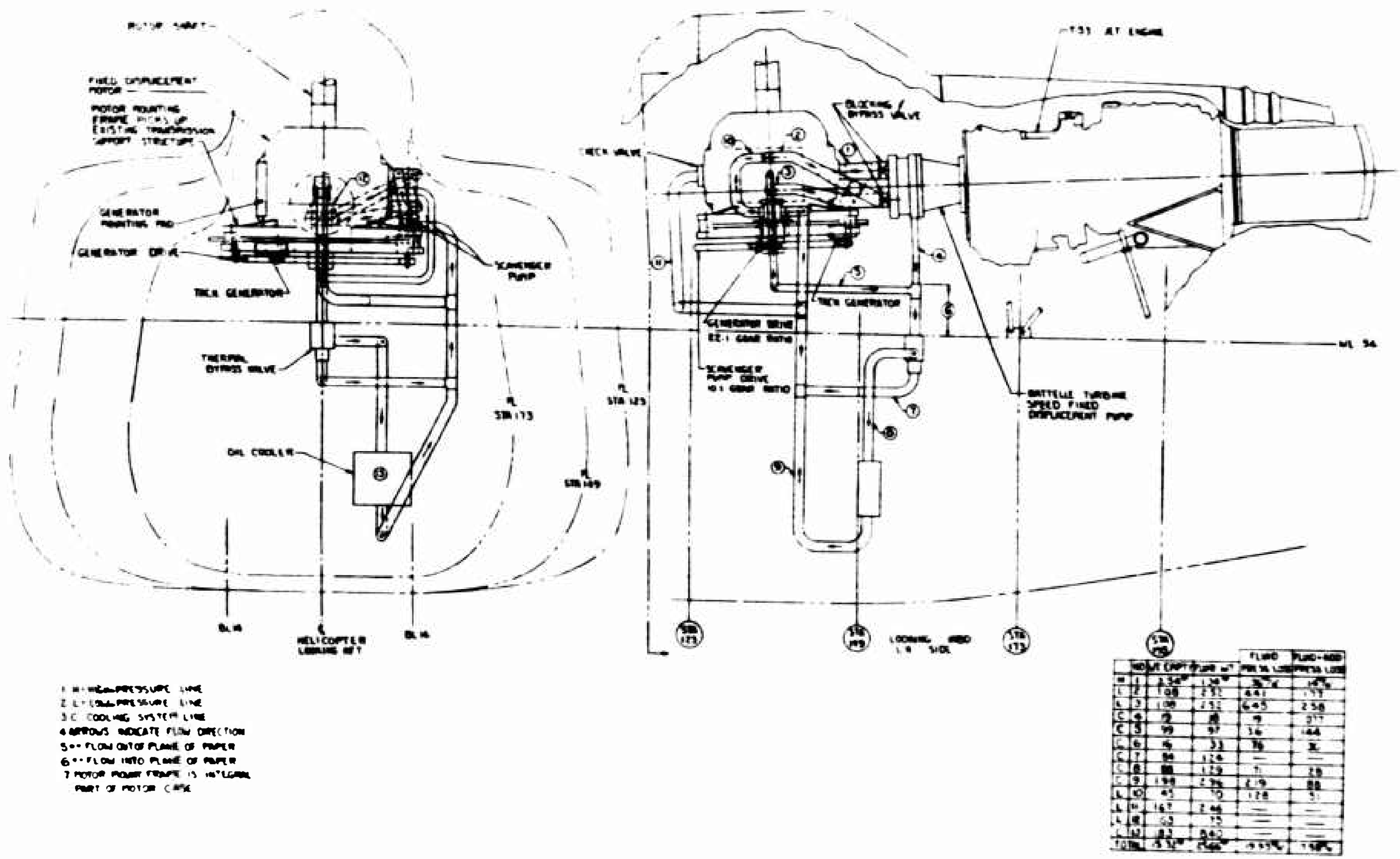

Figure 3. Drafting Layout of Bell UH-1/T-53 Hydraulic Transmission System.

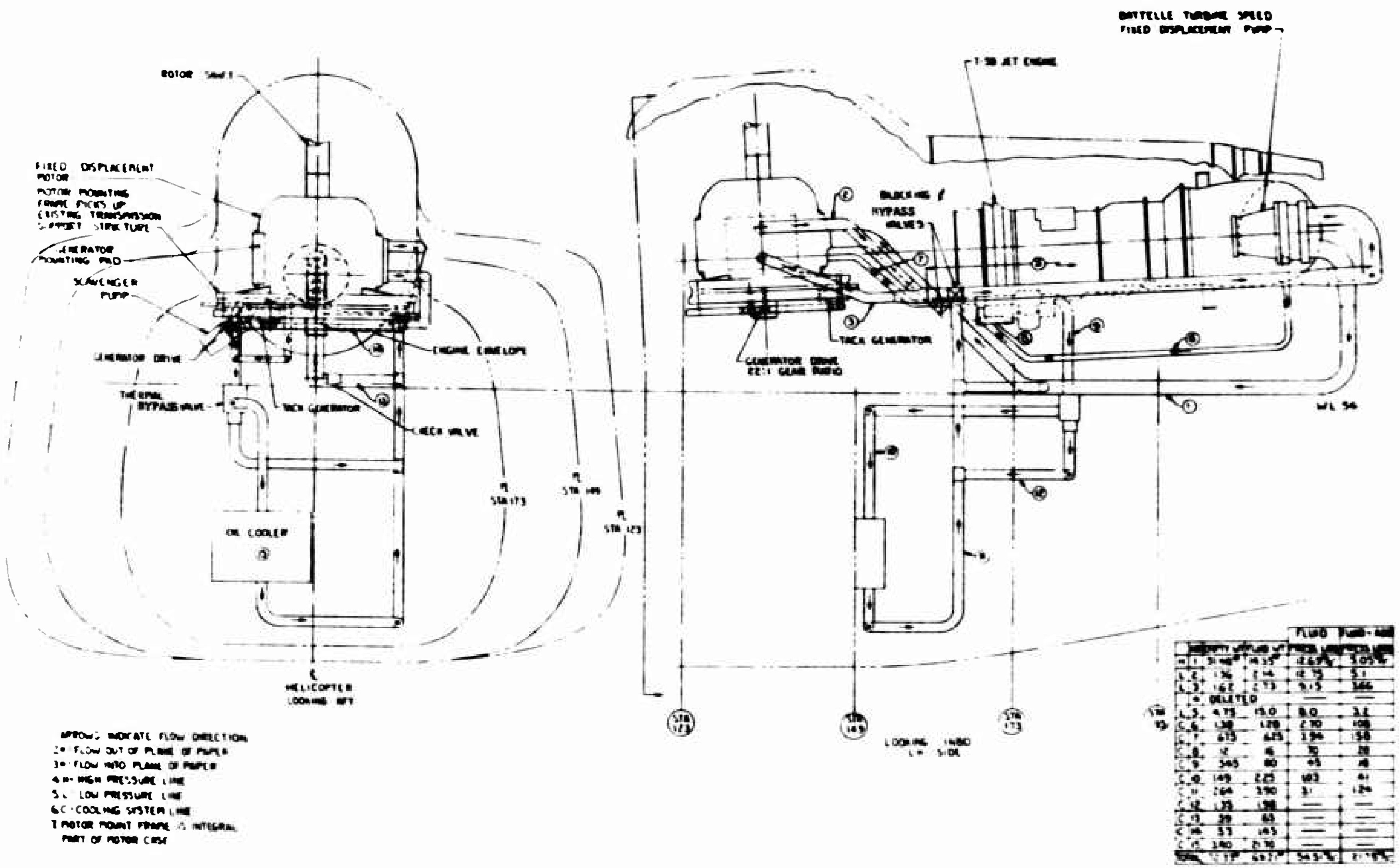

Figure 4. Drafting Layout of Bell UH-1/T-58 Hydraulic Transmission System.

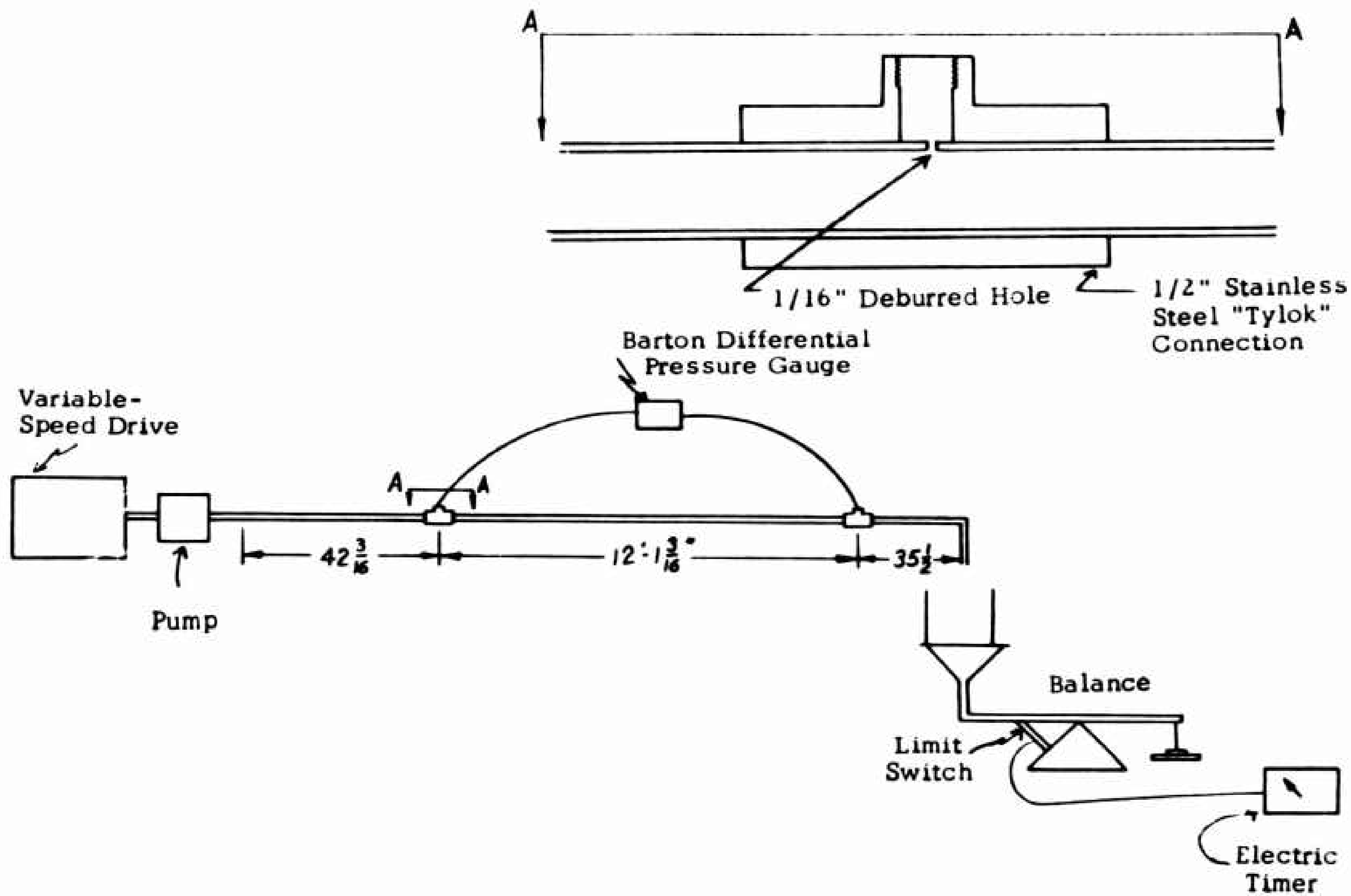

Figure 5. Schematic of 0.416-Inch ID Test Section.

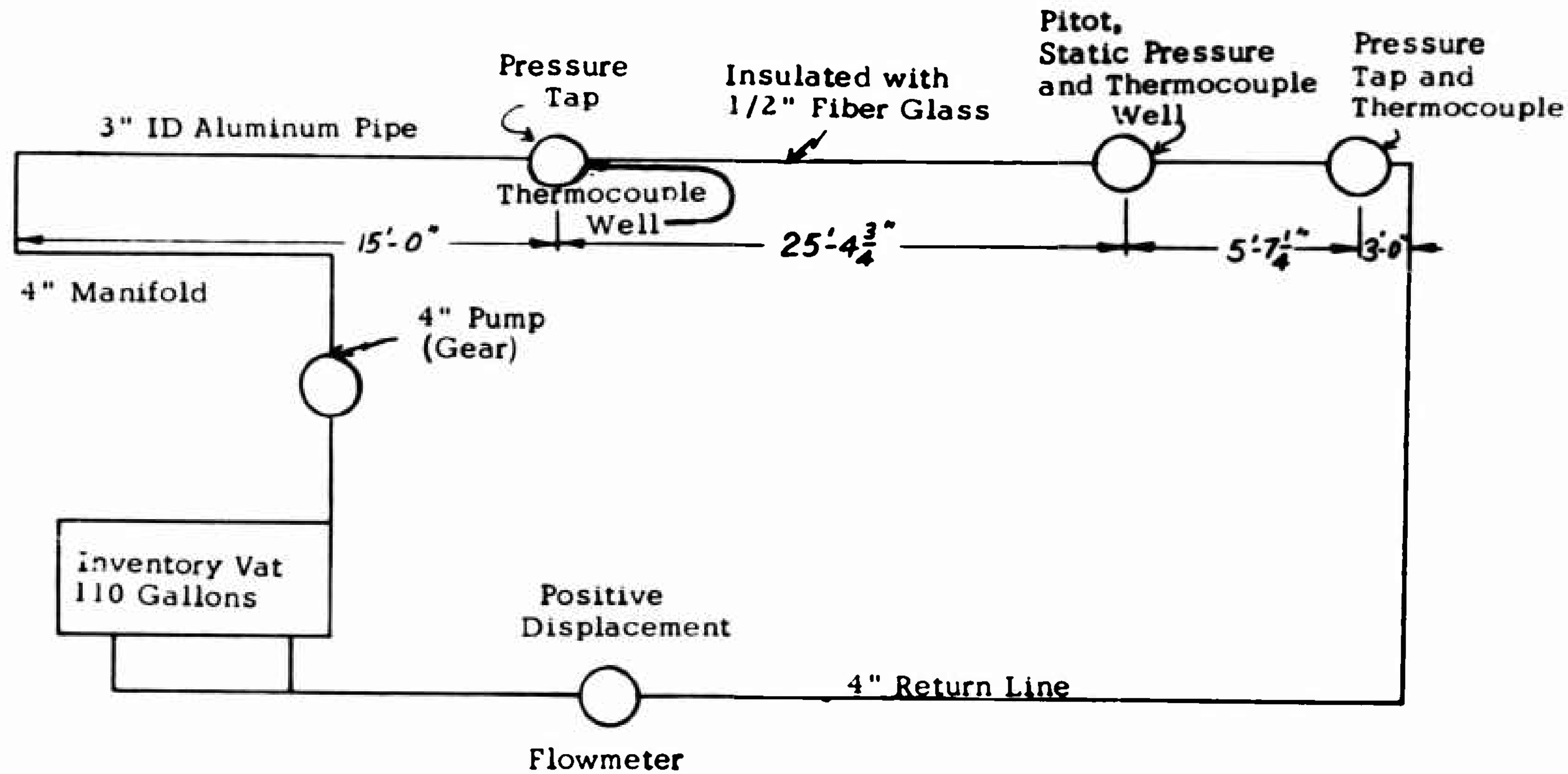

Figure 6. Schematic of 3-Inch Test Section with Location of Pressure Taps, Thermocouples and Pitot Assembly.

MIL-H-5606A With 1.5% By Volume of G-5 + G-15
0.416-Inch ID Tubing (75°F)

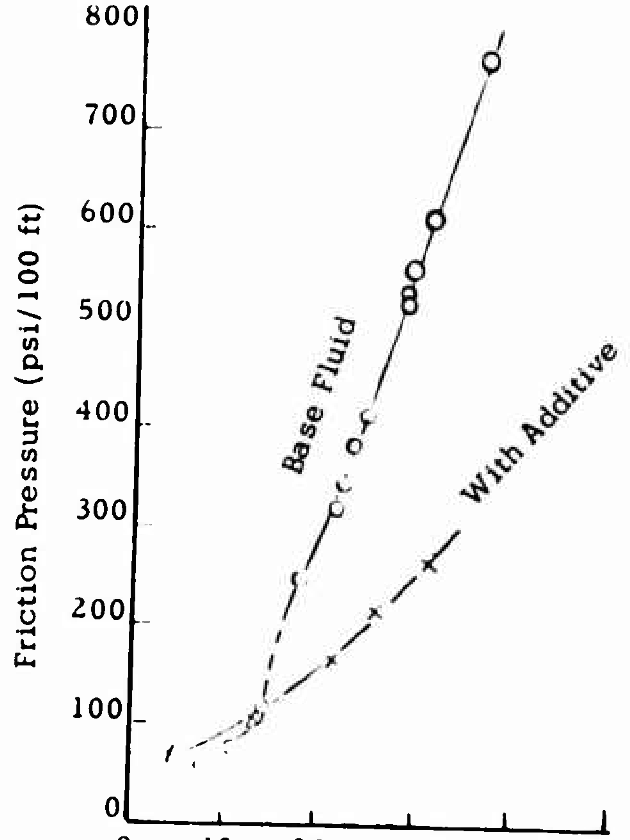

Figure 7. Pipe Friction Pressure Drop Versus Velocity Comparison of Pressure Loss With and Without Improved Fluid Additives.

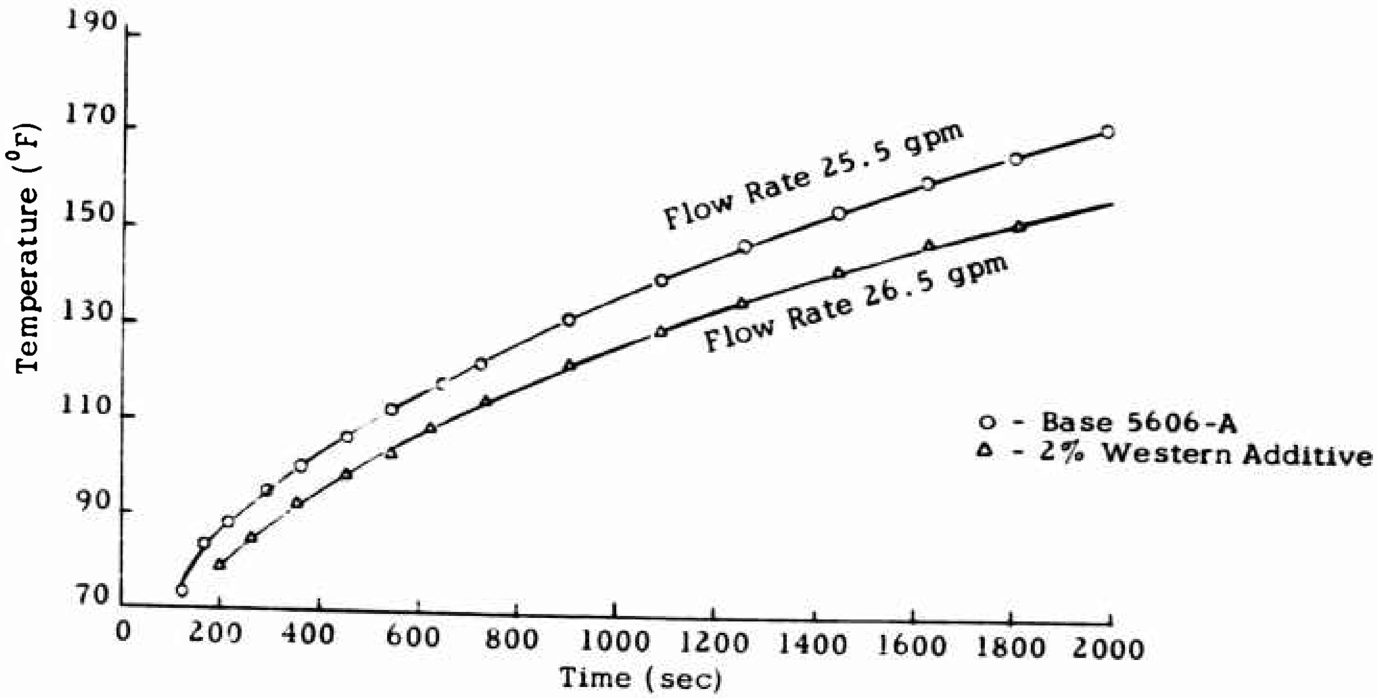

Figure 8. Pump System Temperature Versus Time Comparison of System Temperature With and Without Improved Fluid Additives.

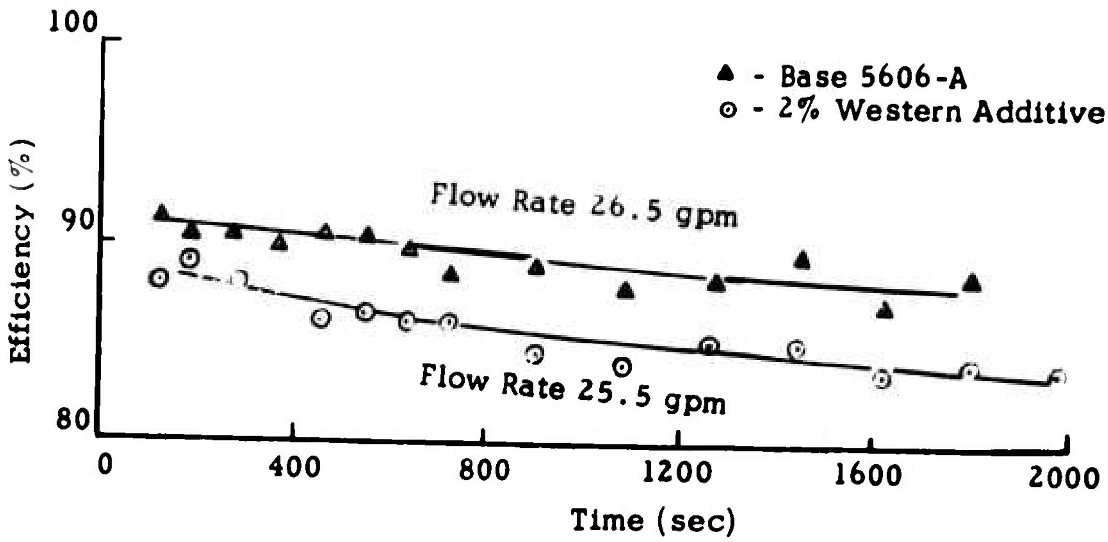

Figure 9. Volumetric Efficiency Versus Time Comparison of Volumetric Efficiency of Pump With and Without Improved Fluid Additives.

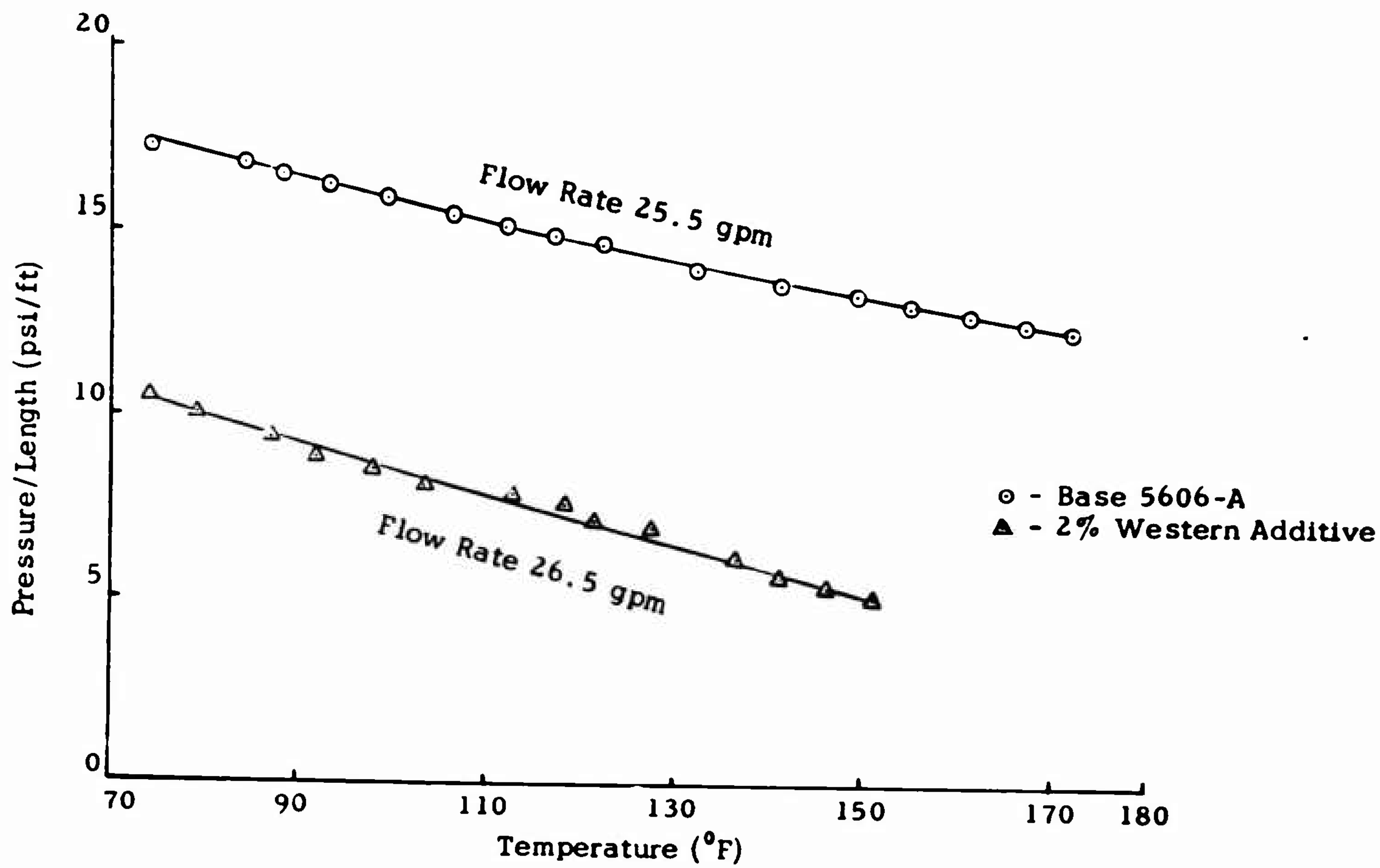

Figure 10. Pipe Friction Pressure Drop Versus System Fluid Temperature With and Without Improved Fluid Additives.

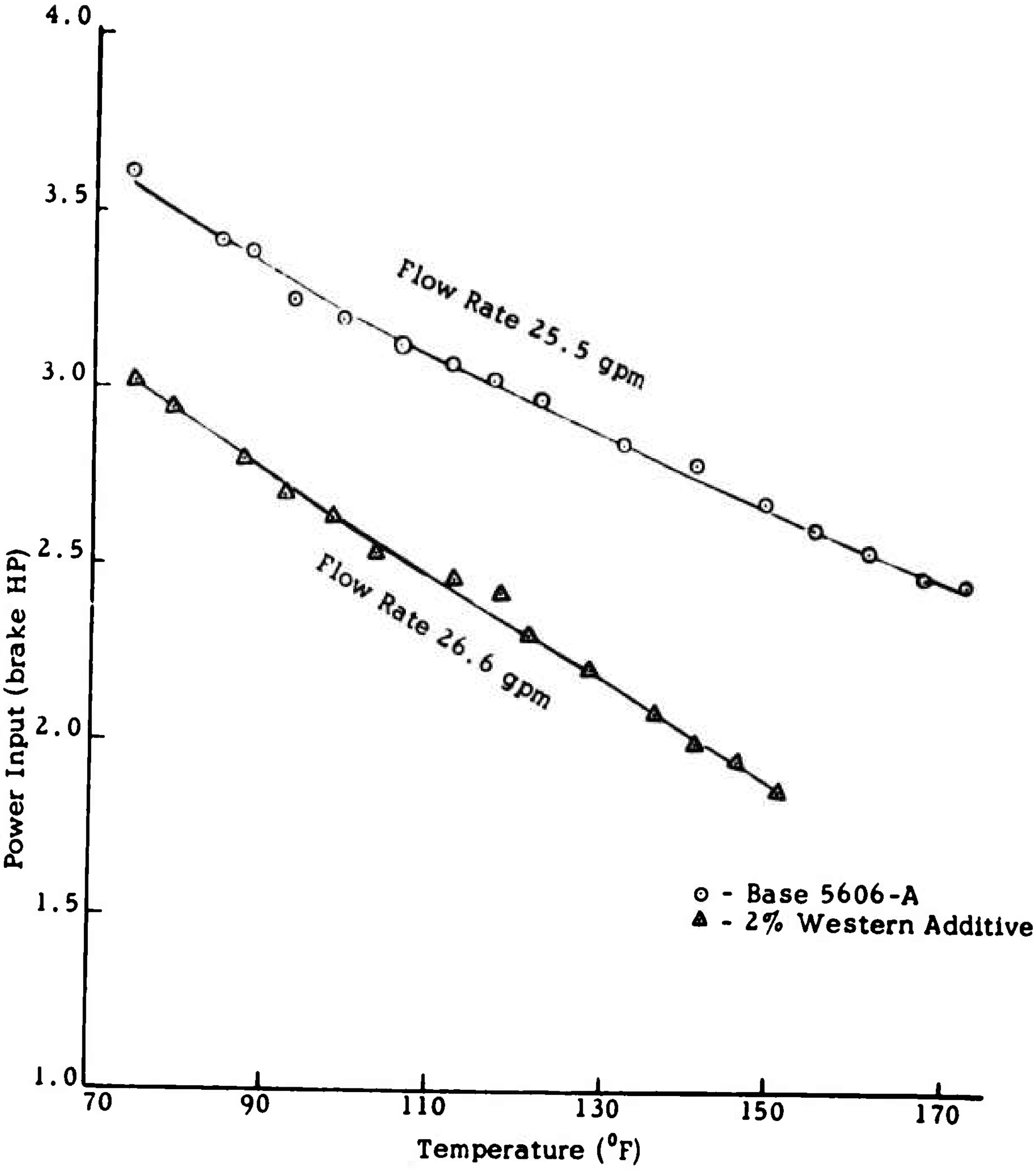

Figure 11. Dynamometer Horsepower to Drive Pump Versus Temperature, Comparison of Pump Horsepower Input With and Without Improved Fluid Additives.

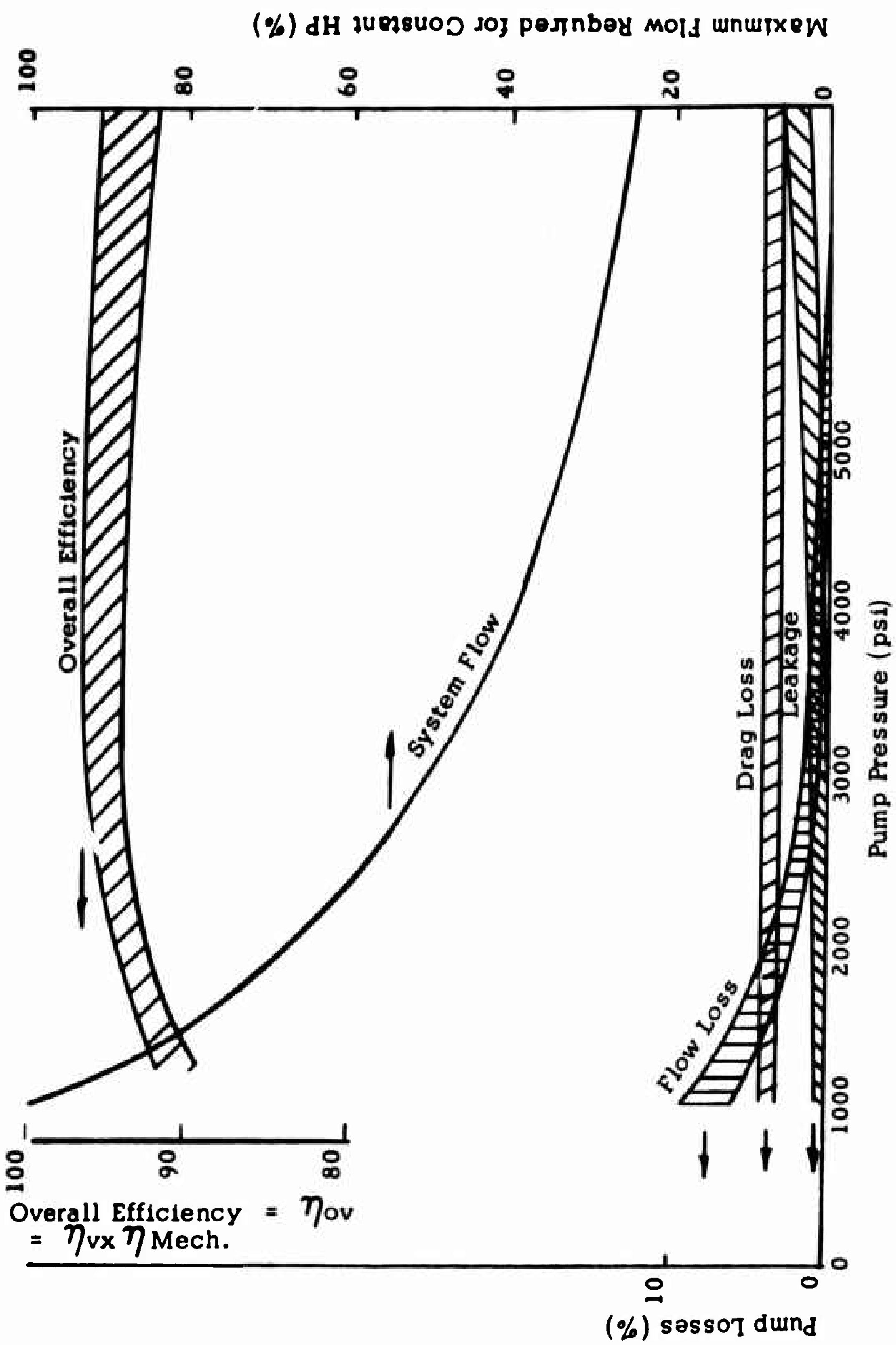

Figure 12. Battelle Fixed-Displacement, Turbine-Speed Pump Performance Versus Pump Operating Pressure.

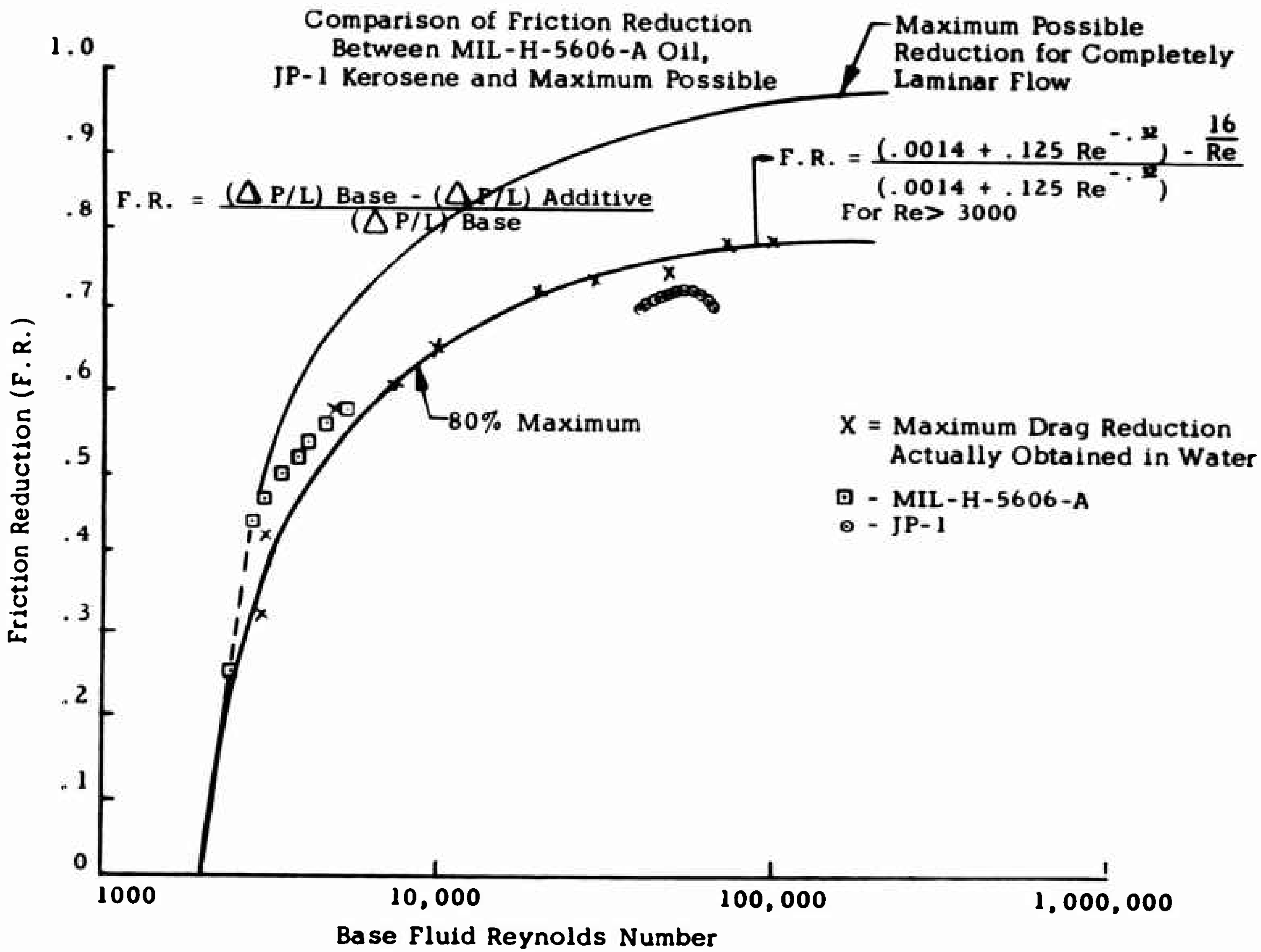

Figure 13. Improved Fluid Friction Reduction Versus Reynolds Number.

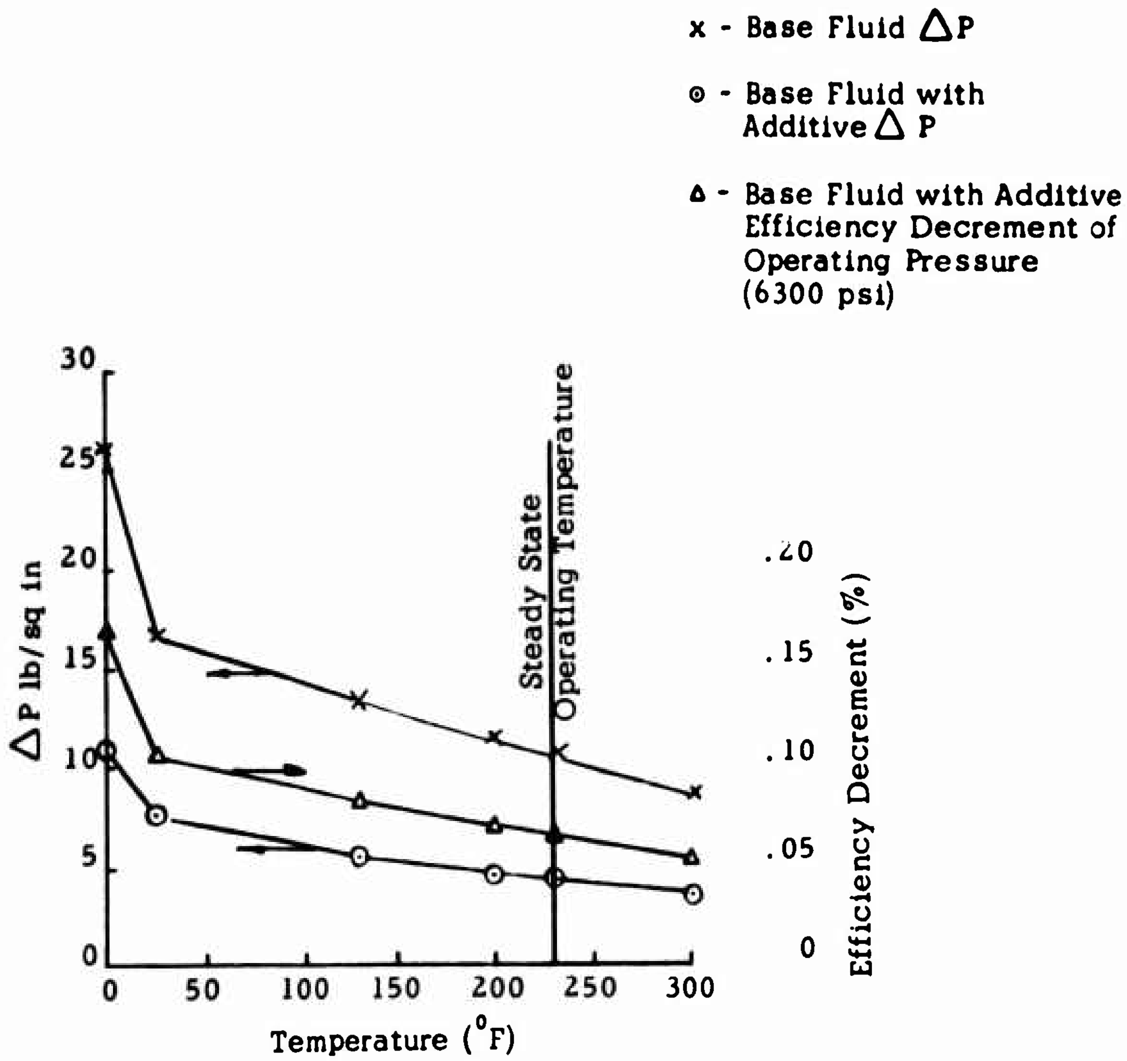

Figure 14. Transmission System Pressure Losses in the Fluid Piping Versus Hydraulic Fluid Operating Temperature, Bell UH-1/T-53 Aircraft.

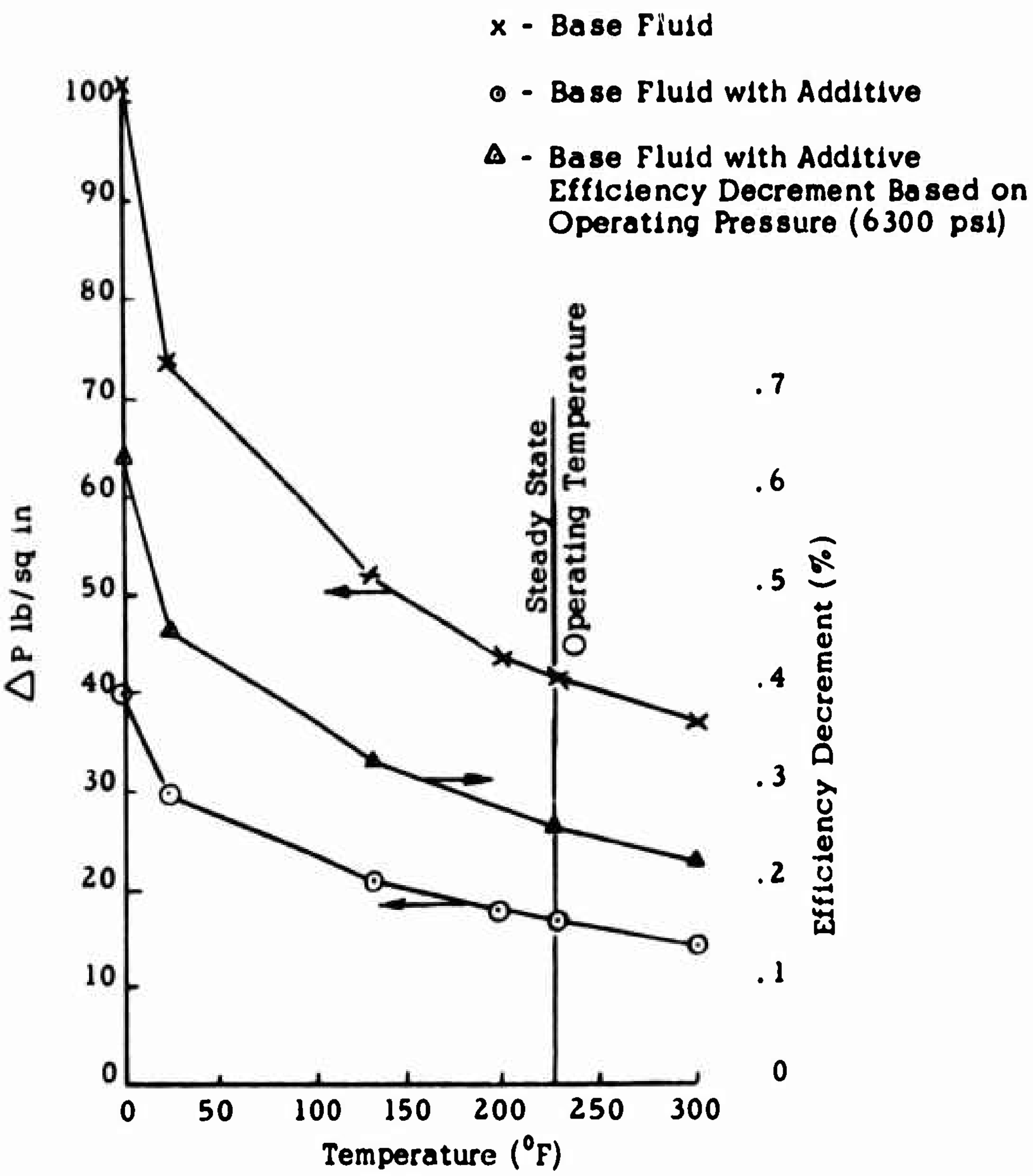

Figure 15. Transmission System Pressure Losses in the Fluid Piping Versus Hydraulic Fluid Operating Temperature, Bell UH-1/T-58 Aircraft.

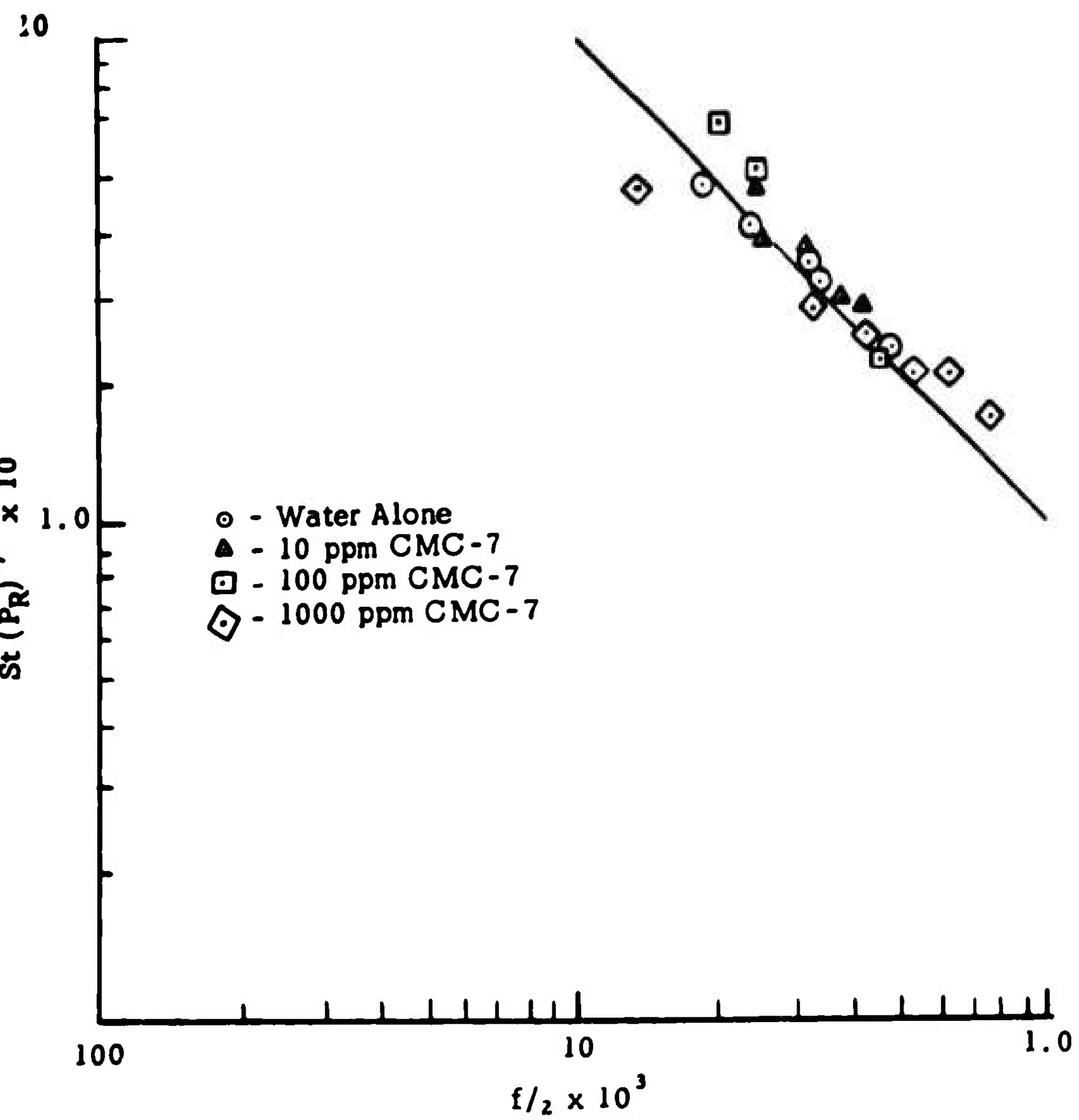

Figure 16. Heat Transfer Correlation for Improved Fluids $S_E(P_R)^{2/3}$ Versus $f/_2$.

Line	Line Weight	Fluid Weight	Fluid Pressure Loss	Fluid + Additive Pressure Loss
High-Pressure Discharge	18.9 lb	4.65 lb	20.4 psi	8.16 psi
Low-Pressure Return	5.45 lb	5.35 lb	16.9 psi	6.8 psi
Totals	24.35 lb	10.00 lb	37.3 psi	14.9 psi

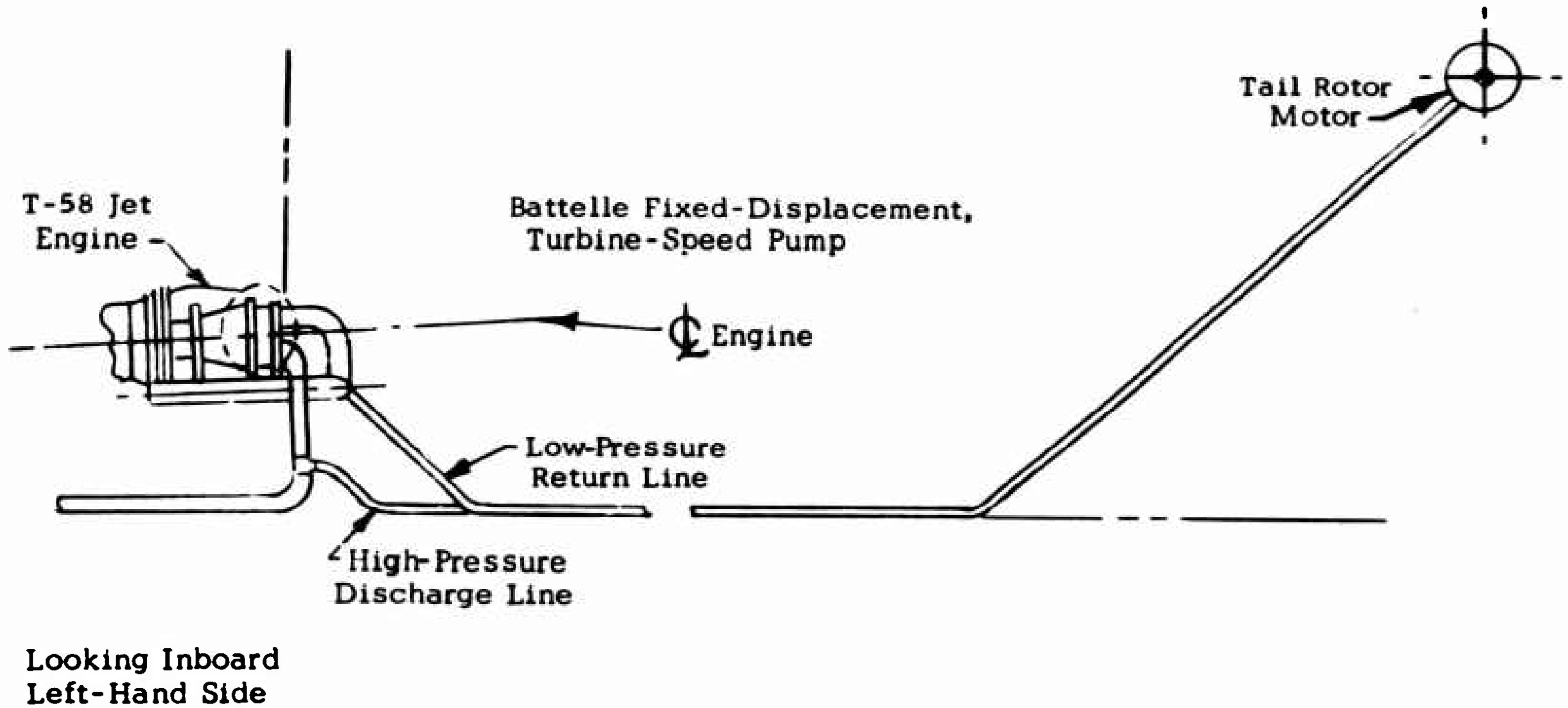

Figure 17. Bell UH-1/T-58 Tail Rotor Hydromechanical Drive System.

Line	Line Weight	Fluid Weight	Fluid Pressure Loss	Fluid + Additive Pressure Loss
High-Pressure Discharge	24.1 lb	5.95 lb	23 psi	9.2 psi
Low-Pressure Return	6.6 lb	6.45 lb	19.5 psi	7.8 psi
Totals	30.8 lb	12.4 lb	42.5 psi	17.0 psi

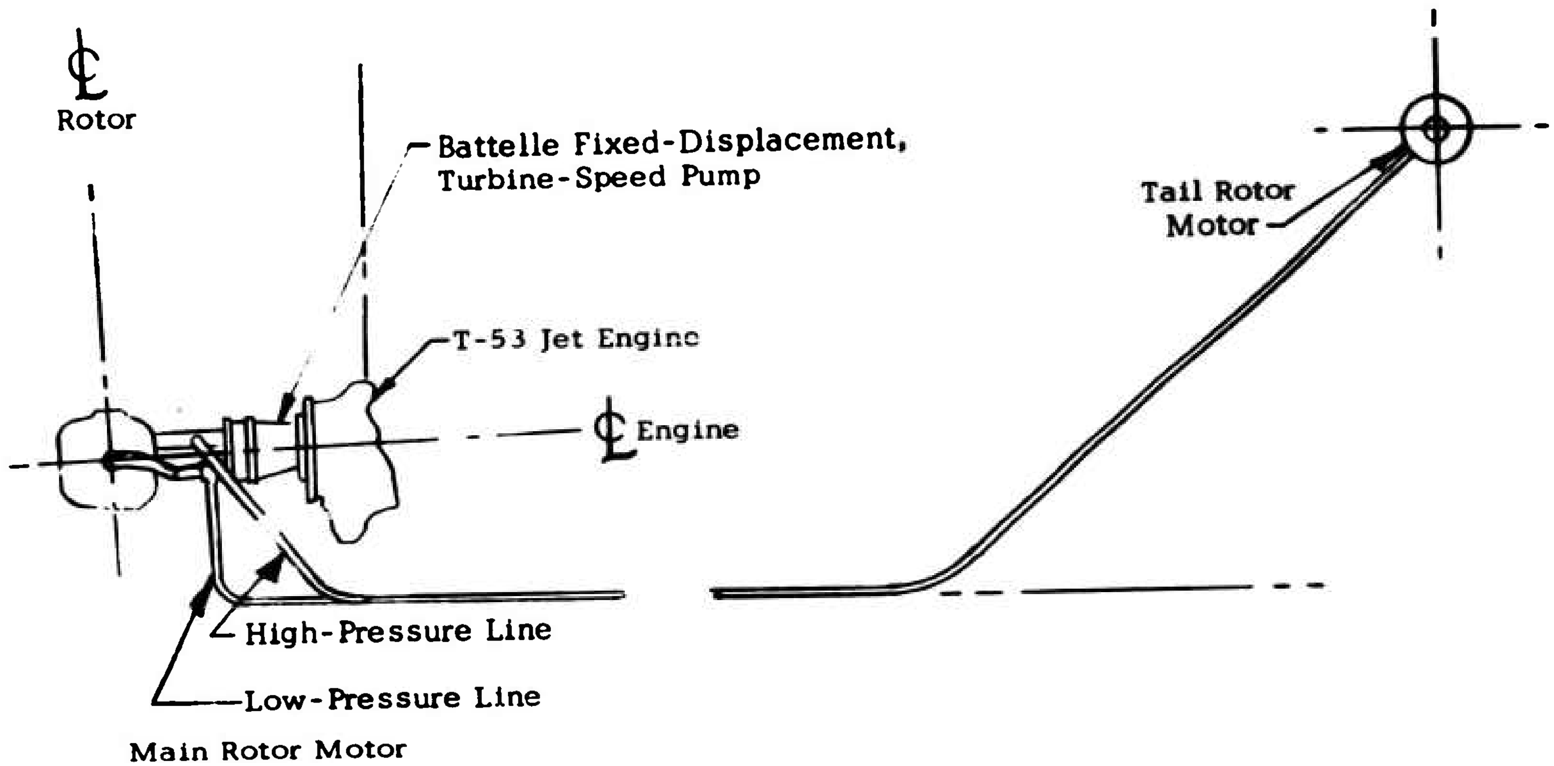

Figure 18. Bell UH-1/T-53 Tail Rotor Hydromechanical Drive System.

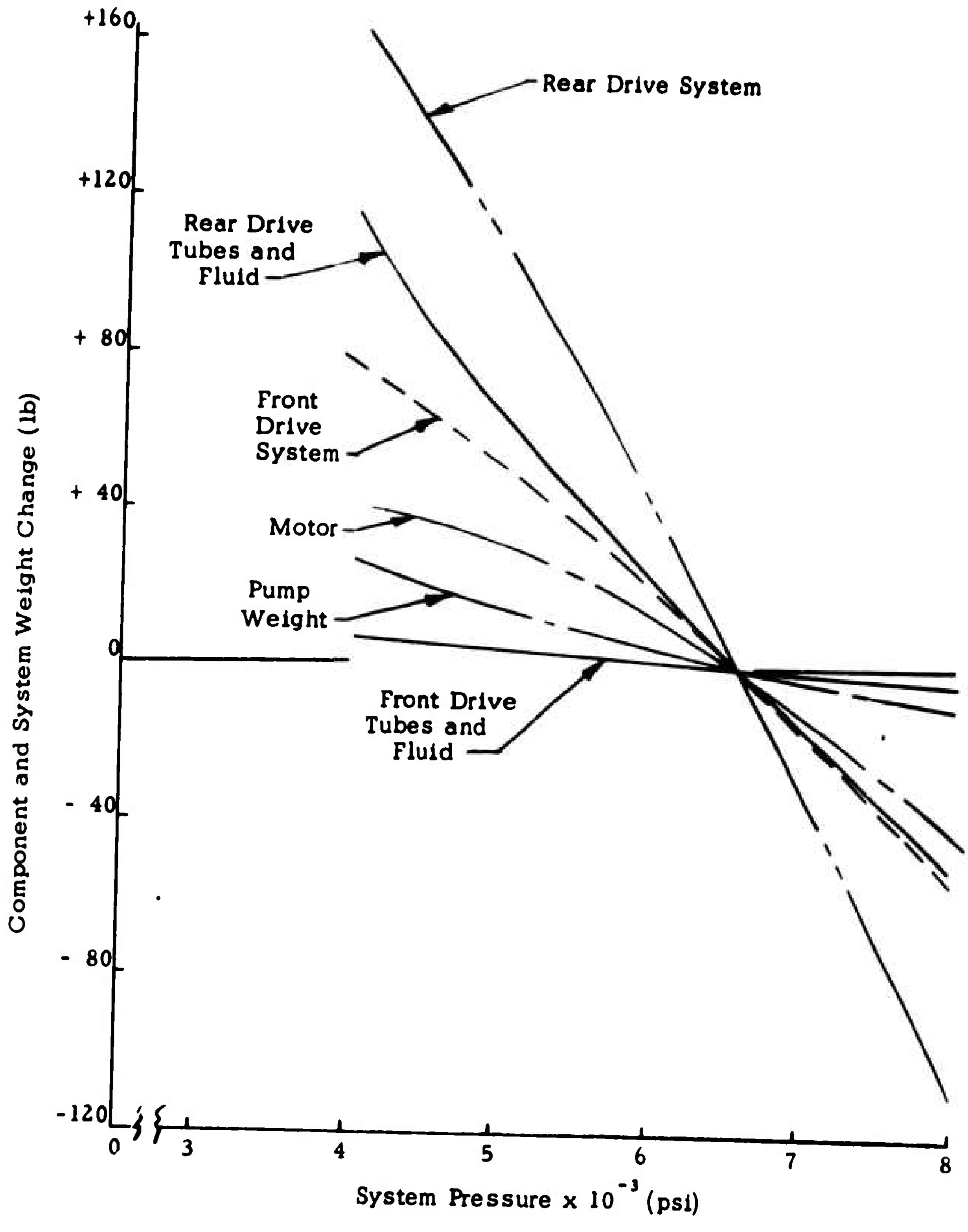

Figure 19. Variation in System Weight Versus System Pressure.

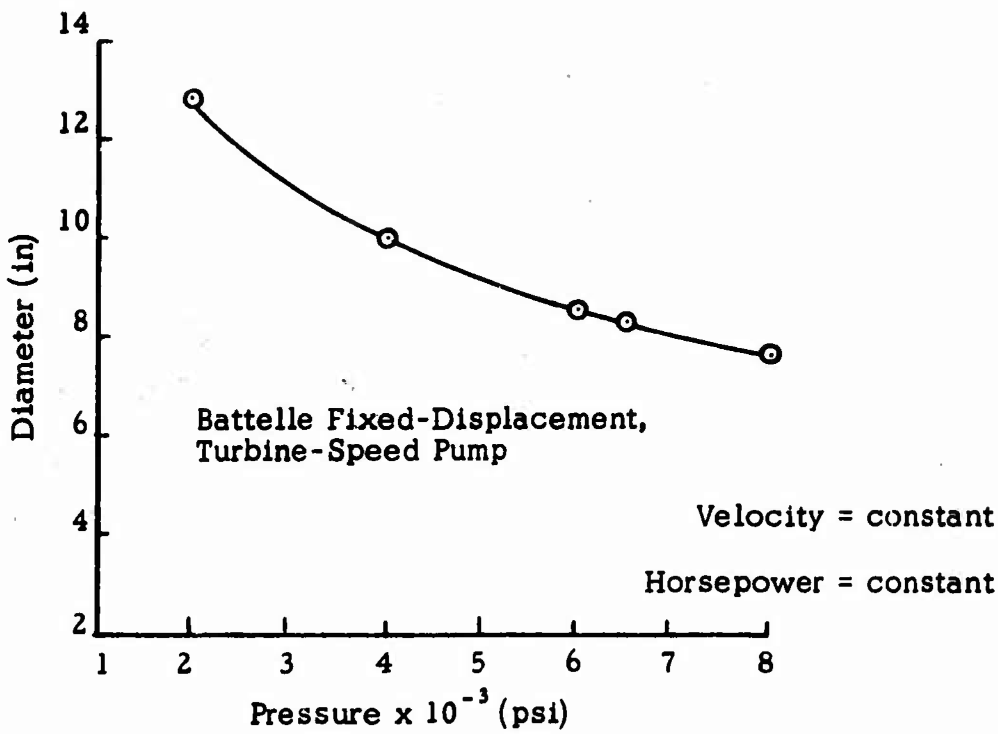

Figure 20. Pump Diameter Versus System Pressure.

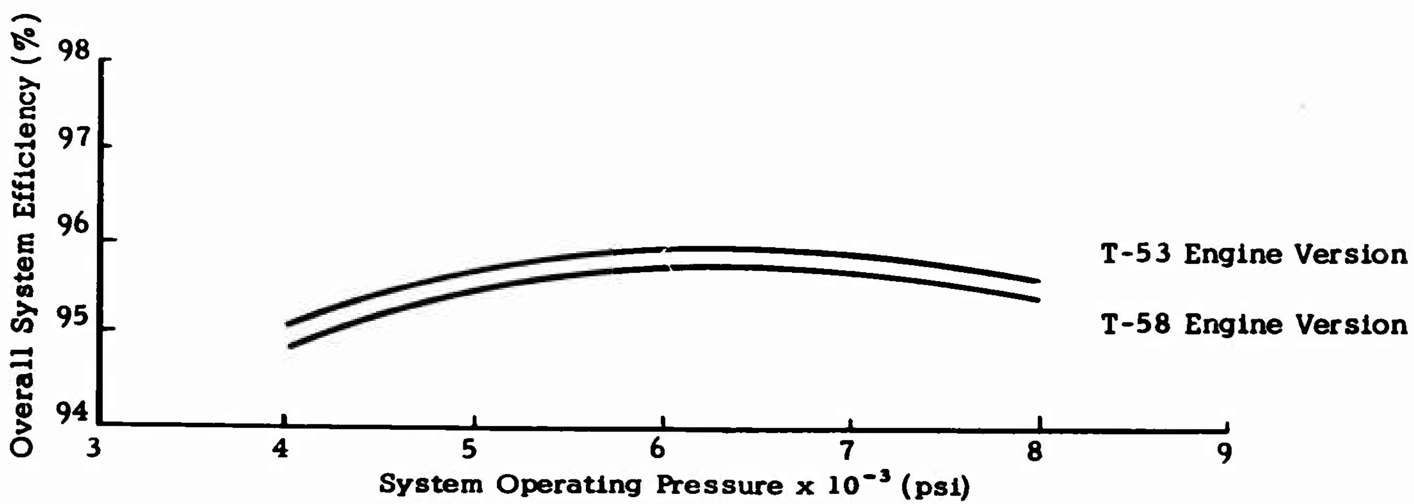

Figure 21. Overall System Efficiency Versus Pressure.

CONCLUSIONS

1. This report indicates that recent advances in the technology of hydraulic pumps and motors and in fluid technology provide a technical breakthrough which could in the near future result in a hydraulic transmission system that is competitive with gear/shaft systems in efficiency and weight.

2. The hydraulic transmission system has inherent characteristics which provide several advantages over the gear transmission system.

3. Consideration of logistics, maintenance, operational and vulnerability problems does not show that the hydraulic system has any major limitations over the gear system.

BIBLIOGRAPHY

1. Brown, G. G., Katz, D., Foust, A. S., and Schnecdewined, R., Unit Operations, John Wiley and Sons, Incorporated, New York, New York, February 1965.

2. Crawford, H. R., and Pruitt, G. T., Investigation for the Use of Additives for the Reduction of Pressure Losses, Final Report for Army Tank Automotive Center Contract No. DA-23-072-AMC-309(T), The Western Company, Richardson, Texas, January 1965.

3. Crawford, H. R., and Pruitt, G. T., A Study of the Mixing of Drag Reduction Polymers, Final Report on Contract No. N123(60530) 34461A, Schedule 40325, Naval Ordnance Test Station, Pasadena, California, The Western Company, Richardson, Texas, October 1964.

4. Kays, W. M., and London, A. L., Compact Heat Exchangers, McGraw-Hill Book Company, New York, New York, 1955.

5. Kreith, F., Principles of Heat Transfer, International Textbook Company, Scranton, New Jersey, 1958.

6. Mitchell, R. K., Thomas, D. L., Swain, J. C., Walowit, J. A., and Wilcox, J. P., Concept and Design Analysis for a Variable Displacement "Turbine Speed" Hydrostatic Pump, Final Report on Contract No. DA-33-019-AMC-378T, Army Tank Automotive Center, Battelle Memorial Institute, Cincinnati, Ohio, May 1965.

7. Perkins, T. K., and Kern, L. R., Widths of Hydraulic Fractures, Atlantic Refining Company, Dallas, Texas, 1963.

8. Schlichting, H., Boundary Layer Theory, McGraw-Hill Book Company, New York, New York, 1960.

9. The Western Company, Investigation of Hydraulic Power Transmission Systems for V/STOL Aircraft, Proposal submitted to USAAML, July 1965.

10. URS Corporation, URS Helicopter Motor, URS Corporation, Burlingame, California, June 2, 1966.

11. Ernst, Walter, Oil Hydraulic Power and Its Industrial Applications, McGraw-Hill Book Company, New York, New York, Second Edition, 1960.

12. Fuller, Dudley D., "Hydrostatic Lubrication," Mechanical Design 19 (Nos. 6, 7, 8, 9), June-September 1947.

13. Ripple, Harry C., Cast Bronze Hydrostatic Bearing Design Manual, Franklin Institute, Philadelphia, Pennsylvania, Cast Bronze Bearing, Incorporated, Cleveland, Ohio, undated.

APPENDIX I

EXPERIMENTAL RESULTS OF IMPROVED HYDRAULIC FLUIDS

TABLE V
PRESSURE DROP AT VARIOUS VELOCITIES IN .416-INCH ID TUBING
BASE LINE CURVE - MIL-H-5606A

Test Section Pressure Drop (psi)	Fluid Flow Time (sec)	Amount of Fluid Flow (cc)	Fluid Velocity (ft/sec)	Pressure Drop/100 ft. (psi/100 ft)
7.1	10.2	2000	8.4	58.7
9.14	15.9	4000	10.8	75.5
13.5	12.0	4000	14.3	112
29.2	11.4	5000	18.8	242
40.5	11.5	6000	22.4	335
93.9	10.7	9000	36.1	776
68.0	11.5	8000	29.8	563
46.5	10.7	6000	24.0	384
66.0	11.7	8000	29.3	546
50.0	12.0	7000	25.0	414
65.4	11.75	8000	29.1	541
38.2	11.9	6000	21.6	316
74.0	11.0	8000	31.1	613

TABLE VI
PUMP AND PIPE PERFORMANCE OF IMPROVED FLUID - MIL-H-5606 WITH 1.5% BY VOLUME OF (G-5 + G-15)

Measured Data			Calculated Results	
12-ft Test Section Pressure Drop (psi)	Fluid Flow Time (sec)	Amount of Fluid Flow (cc)	Fluid Velocity (ft/sec)	Pressure Drop/100 ft (psi/100 ft)
7.9	20.4	2000	4.6	66
12.9	12.35	4000	13.9	107
19.6	7.8	4000	21.9	163
25.95	10.0	6000	25.8	215
31.6	11.1	8000	30.8	262

The above data were taken from the Research Division Test Data Notebook No. 79.

TABLE VII
PUMP AND PIPE PERFORMANCE OF IMPROVED FLUIDS
BASE LINE CURVE IN MIL-H-5606A HYDRAULIC OIL

Total Time from Start (sec)	Corrected Horsepower (HP)	Volumetric Efficiency (%)	Temperature at End of Test Section Normalized to 74°F (°F)	Pressure Drop in Test Section (psi/ft)	Flow Rate (gal/min)
120	3.62	88.0	74	17.25	26.45
180	3.42	89.0	84	16.75	26.8
210	3.38	-	88	16.5	-
280	3.25	88.0	93	16.25	26.45
360	3.20	-	99	15.875	-
450	2.13	86.0	106	15.5	25.9
540	3.07	86.5	112	15.25	26.0
630	3.03	86.0	117	15.0	25.9
720	2.97	86.0	122	14.875	25.9
900	2.85	84.5	132	14.125	25.4
1080	2.79	84.0	141	13.75	25.3
1260	2.68	85.0	149	13.5	25.55
1440	2.61	85.0	155	13.25	25.55
1620	2.54	83.5	161	13.0	25.1
1800	2.47	84.0	167	12.75	25.3
1980	2.45	83.75	172	12.55	25.2

The above are the pump and pipe test data reduced to engineering terms.

TABLE VIII
PUMP AND PIPE PERFORMANCE OF IMPROVED FLUID - RAW TEST DATA

Wt. on 30-in. Torque Arm (grams)	Pressure Drop/ 2 ft (psi)	Temp. in Res. (°F)	Temp. at End of Test Sections (°F)	Amt. of Fluid per Time (gal)	Time for Amt. of Fluid (sec)	Total Time from Start (sec)	Dyna-mometer (rpm)	Air Temp. (°F)
326	0	70	70	0	0	0	0	70.0
2730	34.5	75	74	10	22.7	120	1755	-
2668	33.5	81	84	10	22.4	180	-	-
2650	33.0	84	88	10	-	210	-	-
2581	32.5	90	93	10	22.7	280	-	-
2548	31.75	95	99	10	-	360	-	70.0
2508	31.0	102	106	10	23.2	450	-	-
2480	30.5	108	112	10	23.1	540	1755	70.0
2455	30.0	113	117	10	23.2	630	-	-
2424	29.75	118	122	10	23.2	720	-	70.0
2360	28.25	129	132	10	23.6	900	-	70.5
2324	27.5	138	141	10	23.7	1080	-	70.5
2265	27.0	145	149	10	23.5	1260	-	70.0
2230	26.5	152	155	10	23.5	1440	-	-
2190	26.0	159	161	10	23.9	1620	-	-
2152	25.5	165	167	10	23.7	1800	1750	-
2140	25.1	170	172	10	23.8	1980	-	70.5

TABLE IX
PUMP AND PIPE PERFORMANCE OF IMPROVED FLUID - MIL-H-5606A
WITH 2% BY VOLUME OF (G-5 + G-15)

Total Time from Start (sec)	Corrected Horsepower (HP)	Volumetric Efficiency (%)	Temperature at End of Test Section Normalized to 74°F (°F)	Pressure Drop in Test Section (psi/ft)	Flow Rate (gal/min)
120	3.02	91.2	74	10.5	27.4
180	2.94	90.3	79	10.0	27.15
270	2.8	90.3	87	9.375	27.15
360	2.7	89.8	92	8.75	27.0
450	2.63	90.3	98	8.5	27.15
540	2.53	90.3	103	8.125	27.15
630	2.46	89.6	112.5	7.875	26.9
720	2.42	88.4	118	7.55	26.55
900	2.3	88.8	121	7.1	26.55
1080	2.21	87.6	128	6.75	26.3
1260	2.08	88.1	136	6.15	26.45
1440	1.99	89.4	141	5.75	26.8
1620	1.95	87.2	146	5.4	26.2
1800	1.86	88.4	151	5.125	26.55

TABLE X

PUMP AND PIPE PERFORMANCE OF IMPROVED FLUID - RAW TEST DATA

Wt. on 30-in. Torque Arm (grams)	Pressure Drop/ 2 ft (psi)	Temp. in Res. (°F)	Temp. at End of Test Sections (°F)	Amt. of Fluid per Time (gal)	Time for Amt. of Fluid (sec)	Total Time from Start (sec)	Dyna-mometer (rpm)	Air Temp. (°F)
362	0	74	75	0	0	0	0	70.0
2454	21.0	76	79	10	21.9	120	1755	70.0
2410	20.0	81	84	10	22.1	180	-	-
2330	18.75	87	91	10	22.1	270	-	-
2276	17.5	93	97	10	22.2	360	1755	-
2238	17.0	99.5	103	10	22.1	450	-	-
2185	16.25	105	108	10	22.1	540	-	-
2150	15.75	110	113	10	22.3	630	1755	69.0
2125	15.1	114	117.5	10	22.6	720	-	-
2065	14.2	123	126	10	22.5	900	-	-
2015	13.5	131	133	10	22.8	1080	-	-
1940	12.3	138	141	10	22.7	1260	-	-
1905	11.5	144	146	10	22.4	1440	-	-
1870	10.8	149	151	10	22.9	1620	1755	70.0
1825	10.25	154	156	10	22.6	1800	-	-
1800	10.0	158	160	10	22.9	1980	-	-

The data in Tables IX and X were taken from Research Division Laboratory Notebook No. 51.

ENDURANCE TESTING

A 1.5 percent by volume of G-5 + G-15 mixture was combined with MIL-H-5606A and tested in the test apparatus shown in Figure 6. The test apparatus was operated six to eight hours a day during the work days of the week. Two and one-half months after the beginning of these tests, in excess of 200 hours had been accumulated on this test apparatus with the original oil. The oil was circulated at 360 gallons per minute in this closed-loop system. These data are recorded in the Research Division Notebook 79. The oil from this test apparatus was removed and stored when the test apparatus was used for other testing. After a period of storage, the oil was put back into the test apparatus and additional endurance testing was performed. No degradation in friction reduction performance was detected during endurance tests.

APPENDIX II

BATTELLE FIXED-DISPLACEMENT, TURBINE-SPEED PUMP SIZING AND PERFORMANCE

1. Listed below are the significant design parameters of the Battelle Fixed-Displacement, Turbine-Speed Pump. (The calculated results were included with the sizing study performed by Battelle. A pump drawing has also been supplied by Battelle.)

   a. Pump is a fixed-displacement pump.
   b. Designed for an input horsepower of 1500.
   c. Sized to operate at 20,960 rpm with allowable 2-second overshoots to 21,275 rpm (same as maximum engine operation).
   d. Operating pressure of 6600 psi.
   e. 200°F allowable ambient operating temperature.
   f. Oil viscosity = 6 centistokes at 300°F.
   g. Volumetric efficiency = 98% (with existing hydraulic fluid).
   h. Mechanical efficiency = 97.5% (with existing hydraulic fluid).
   i. Overall efficiency = 95.5% (with existing hydraulic fluid).

   The 2.5 percent of mechanical deficiency consists of 2-percent fluid flow losses and .5-percent mechanical drag or viscous shear.

   j. Weight of pump = 70 pounds.

A detailed technical review of the pump efficiency and weight was performed by The Western Company to verify calculated pump weight and efficiency. The calculation techniques and test results shown in Reference 6 were used to verify efficiency.

2. Listed below is the performance of the Battelle Fixed-Displacement Pump with the improved fluid at the design rated point of 6600 psi and 1500 horsepower and rated speed.

   a. Mechanical efficiency = 98.7%
      (1) Fluid flow losses = 0.08%
      (2) Viscous shear losses = 0.5%

   b. Volumetric efficiency = 98%

   c. Overall efficiency = 96.7%

3. Listed below is the performance of the Battelle Fixed-Displacement Pump with the improved fluid at 50-percent power at rated speed, 3300 psi and 750 horsepower.

   a. Mechanical efficiency = 97.4%

   b. Volumetric efficiency = 99.5%

c. Overall efficiency = 96.9%

4. The basis for pump performance is the Battelle report cited as Reference 6. The following excerpts from that report are descriptive of the techniques used to establish pump performance.

Page 15.

"A band of values (performance) is shown for each of the major losses. In general, our calculations predict the lower values for the losses and consequently the high value for the pump efficiency. The width of the bands of losses therefore represents a conservative allowance. The losses were arrived at as follows:

DRAG The drag losses include

- Vane tip drag
- Drag of the rotor
- Drag of the fluid between the vanes

The largest single element of drag is the vane tip drag. The value assigned for it is an average of the computer analysis and vane tip experiment results. The calculations show a total drag loss of 3.1 percent of total horsepower and the band was set at 3.1 to 4.1 percent in the prediction. This loss will be essentially constant at a constant pump speed.

FLOW LOSSES The band of losses shown is based strictly on the pressure-drop experiments. The lower value represents the loss to be expected with 60-percent open flow area and the upper value for 50-percent open flow area. The losses in the inlet and exhaust manifolds will be negligible.

LEAKAGE Both the computer and hand calculations predict a very low value of leakage even at the high pressure end of the constant-power range. The calculated value of about 1 percent has been arbitrarily tripled to allow for the distortion of parts, manufacturing tolerances, and other factors which may arise in a 8000-psi pump. One of the principal advantages of a high-speed pump is, of course, its low leakage at high pressure. It will remain to be proven whether it is as low as the calculations suggest."

"This project work is recorded in Battelle Laboratory Record Books 21520, 28126, 21764, and 21669."

As seen from the above explanation, most of the performance characteristics of the pump come from experimental work or were checked by experimental results.

APPENDIX III

DETAILED DESCRIPTION OF METHODS FOR CALCULATING LOSSES IN THE URS MOTOR

There are seven points at which significant torque (fluid or mechanical friction) or volumetric losses occur in the URS motor. These are tabulated below:

Loss Number	Description	Type of Loss
1	Piston Bearing Pad	Volumetric and Fluid Friction
2	Ball	Mechanical Friction
3	Valve Journal	Mechanical Friction
4	Main Bearing	Mechanical Friction
5	Fluid Flow and Acceleration	Fluid Friction
6	Piston	Volumetric and Fluid Friction
7	Valve	Volumetric and Fluid Friction

Not included are losses due to scavenging and cooling pumps (which serve both the main pump and the motor), residual oil in the crankcase (it is intended to operate the unit with a dry sump), fitting at inlet and outlet of the motor, shaft seal, and main bearing preloads. Most of these are charged to system losses. The remainder are negligible.

The methods by which the losses shown in the table were calculated are given in detail in the following material.

Loss Number 1, Bearing Pad

One of the more critical elements in the URS design is the use of the specially designed hydrostatic bearing pad between the piston and the shaft-mounted, rotating, variable eccentric. As noted by Ernst (Reference 11), "Heavy thrust forces occurring in hydraulic pumps and motors can be carried by the hydrostatic oil films without metal-to-metal contact at minimum friction."

Professor Dudley D. Fuller, Chairman of the Department of Mechanical Engineering, Columbia University, who has written extensively on the subject (Reference 12), and his associate Stanley Abramovitz, formerly in charge of the lubrication group at the Franklin Institute, were retained for

basic design and analysis of this bearing. The procedures for calculating losses as recommended by these consultants and employed by them in the initial calculations involve trial and error determination of the mutual effect of oil temperature and viscosity (i.e., the reduction in viscosity due to temperature rise and the reduction in temperature due to viscosity decrease). Because of the importance of the bearing pad in the URS design, and because the temperature-viscosity interplay could best be found with high-speed iterative techniques, a computer program has been prepared for this analysis. The procedures to be described are, however, essentially those employed in the initial hand calculations and have been checked by Abramovitz.

The procedure begins with a choice of basic bearing configuration. (The four-recess, orifice-fed configuration was recommended by Fuller and Abramovitz because of its ability to compensate for cocking forces in any direction.) Pad coefficients are then found from plotted values derived with an analog computer at the Franklin Institute (Reference 13). With these coefficients, the pad recess pressure required to maintain an oil film under the load generated by pressure on the upper surface of the piston is found. Then, at the lowest speed and highest pressure for maximum horsepower operation, the computer calculates the size of orifice (that orifice which feeds a pad recess) which will pass the flow required to maintain a minimum (chosen) clearance between pad and eccentric. With this--now fixed--orifice size, the computer then calculates the clearances that would exist at the highest designed motor speed and at 30 other speeds between zero and the maximum. At each of these speeds, the computer also calculates losses due to flow through the orifice, HP_0,

$$HP_0 = \frac{Q\,t_0}{k_1} = \frac{Q(P_s - P_r)}{k_2} \qquad (5)$$

where

Q	=	flow through the orifice
t_0	=	temperature rise across the orifice
P_s	=	supply (cylinder) pressure
P_r	=	recess pressure
k_1 and k_2	=	constants of proportionality

the losses due to extrusion out the pad, HP_e,

$$HP_e = \frac{QP_r}{k_2} \qquad (6)$$

and the losses due to shear (friction) of the fluid between pad and eccentric, HP_s,

$$HP_s = \frac{\mu_{tot}\, a_{sill} (V_{av})^2}{hk_3} \tag{7}$$

where

μ_{tot} = the viscosity of the fluid at the appropriate temperature

a_{sill} = the sill area of the bearing pad

V_{av} = the rubbing speed of the pad-eccentric interface

h = the radical clearance of the pad from the eccentric

k_3 = a constant of proportionality

Fluid temperature rise is also calculated at each step, and indeed--through an iterative process--the computer finds the effect of temperature rise on viscosity. Then the new temperature rise caused by the new viscosity (and therefore losses) is obtained. The extrusion loss is a volumetric loss; the orifice and shear losses are considered as fluid friction losses.

Loss Number 2, The Ball

The cylinder ball is a second important feature contributing to the significant improvement of the URS unit over more conventional units. Its basic function is to enable the elimination of the greatest part of piston side loading. The only side loads remaining are a small oscillating load due largely to inertia of the ball and friction at its seats and a small load transmitted through fluid shear at the bearing pad.

Loss at the ball is caused by friction due to the ball's small oscillating motion under seat load forces only (the ball itself is balanced against system pressure). Because of the ball's small motion, full fluid film lubrication is not required and fluid losses under boundary lubrication are negligible. The friction loss at the ball seat HP_B is calculated in the same manner as for a shaft seal (except note again that the motions are very small) and is given (by Abramovitz), HP_B,

$$HP_B = \frac{4\alpha TN}{360k_4} \tag{8}$$

where

α = the angle through which the ball rotates during one revolution

T = the frictional torque produced by the ball

N = the motor speed

k_4 = a constant of proportionality

This is a mechanical friction loss.

Loss Number 3, Valve Journal

The valve journal bearing is externally lubricated (with oil pressure supplied by the external cooling pump). The bearing operates under very low load (since the valves are completely balanced); therefore, relationships for concentric bearings are applicable. From Fuller's book (pages 227-229 of Reference 12), the frictional moment for such a bearing, M,

$$M = \frac{k^5 \mu \ell r^2 N}{m} \tag{9}$$

where

μ = viscosity

ℓ = the axial length of the bearing

r = the bearing radius

N = the rotational speed

m = the clearance modulus = the radial clearance/r

and the loss in such a bearing, HP_j,

$$HP_j = \frac{MN}{k_6} \tag{10}$$

where

k_5 and k_6 are constants of proportionality.

This is a mechanical friction loss.

Loss Number 4, Main Bearing

There are, of course, no thrust forces on the main bearing. Techniques for calculating this loss were drawn from the SKF Engineering Journal, from which a value of 0.0018 was taken as the friction coefficient.

The relationship employed, HP_{MB},

$$HP_{MB} = \frac{T \times N \times R\,(.0018)}{k_7 \times S/2} \tag{11}$$

where

T = torque

N = rational speed

R = bearing radius

S = stroke

k_7 = a constant of proportionality

This is a mechanical friction loss.

Loss Number 5, Fluid Flow and Acceleration

Calculations were made for losses in the orifices between cylinder and passage (5 consecutive orifices were taken), losses in the passages themselves, and losses due to accelerating and decelerating the fluid. A conservative orifice coefficient of 0.6 was assumed.

Orifice losses, HP_{FO},

$$HP_{FO} = \frac{5Q\rho V^2}{k_8 c_d^2} \tag{12}$$

where

Q = flow

ρ = fluid density

V = fluid velocity

c_d = orifice coefficient

Passage losses, HP_{FP},

$$HP_{FP} = \frac{QfLV^2 s}{k_9 D} \tag{13}$$

where

f = friction factor

L = passage length

V = fluid velocity

s = fluid specific gravity

Acceleration losses, HP_{FA},

$$HP_{FA} = \frac{QVS}{k_{10}Ng} \tag{14}$$

where

S = stroke

N = rotational speed

g = acceleration of gravity

k_8, k_9, k_{10} = constants of proportionality

These are fluid friction losses.

Loss Number 6, Piston

This loss consists of leakage past the piston as well as oil shear or friction produced by the sliding of the piston within the bore. The formulae for this calculation are those customarily used for calculation of piston losses in any pump and as a reference are given by Ernst (pages 42-52 of Reference 11). Formulae 5.18, 5.37, and 5.42 were combined to take the most conservative values produced due to a relative motion of the piston within the bore, as well as its possible eccentricity within the bore. This latter value was taken at a maximum which cannot occur in practice and was taken only as a limiting condition of the most conservative calculation. For the relationship used, see the discussion of valve loss below. This is both a volumetric and a fluid friction loss.

Loss Number 7, Valve

This represents the leakage and shear or fluid friction at the valve, and has been calculated separately for the short lands and for the long lands. The same formula as that for the pistons is used because the conditions are the same except for the length of land, which is calculated separately for each instance. The variation for the length of the short land was accounted for as it describes its stroke.

$$HP_{VP} = \frac{nD}{k_{11}} \quad \frac{\nu S^2 \pi L}{k_{12} b} + \frac{k_{13} P^2 b^3}{\nu L} \left(1 + \frac{3}{2} \epsilon^2\right) + \frac{k_{14} s b}{2} \tag{15}$$

where

n	=	the number of elements producing the loss
D	=	the diameter of the valve or piston
ν	=	the viscosity of the fluid
S	=	the average valve or piston speed
L	=	the length of the element
b	=	the radial clearance
P	=	the pressure of the fluid
ϵ	=	the eccentricity of the valve or piston in the bore
k_{11}, k_{12}, k_{13}, k_{14}	=	constants of proportionality

This is both a volumetric and a fluid friction loss.

TABLE XI
TABULATION OF DETAILED LOSSES IN MOTOR AT 300 RPM

	Horsepower	Percentage
Bearing pad losses $HP_0 + HP_e + HP_s$	8.860	.594
Ball losses HP_B	.368	.025
Valve journal losses HP_j	.247	.016
Main bearing losses HP_{MB}	.081	.0054
Fluid flow and acceleration $HP_{FO} + HP_{FP} + HP_{FA}$	6.520	.435
Piston losses HP_P	2.900	.194

TABLE XI - Contd.		
	Horsepower	Percentage
Valve losses HP_{VP}	4.430	.297
TOTAL	23.406	1.57
Overall efficiency	98.43	
Design point - 6600 psi, 1500 horsepower		

TABLE XII

TABULATION OF DETAILED LOSSES IN MOTOR AT 50% POWER

	Horsepower	Percentage
Bearing pad losses $HP_0 + HP_e + HP_s$	8.640	1.15
Ball losses HP_B	.368	.49
Valve journal losses HP_j	.247	.33
Main bearing losses HP_{MB}	.081	.011
Fluid flow and acceleration $HP_{FO} + HP_{FP} + HP_{FA}$	6.520	.87
Piston losses HP_P	2.350	.31
Valve losses HP_{VP}	4.100	.53
TOTAL	22.306	2.97
Overall efficiency	97.03	
50% power - 3300 psi, 750 horsepower		

APPENDIX IV

HYDRAULIC TRANSMISSION HEAT EXCHANGER SIZING

The heat exchanger parameters for the proposed hydromechanical transmission will be obtained by scaling up the existing transmission heat exchanger to account for the differences in the two systems.

The differences between the hydraulic system and the existing transmission heat exchanger are listed below.

		Hydraulic	Gear Transmission
1.	Horsepower	1500	1375
2.	Efficiency decrement	5.0	2.2[#]
3.	Heat transfer coefficient		.52

1. Horsepower and Efficiency Effects on Heat Exchanger

It is seen from the heat exchanger equation that it is possible to scale up the heat exchanger by increasing the exchanger area.

$$Q = U_A A_A (T_{A_2} - T_{A_1}) = U_0 A_0 (T_{02} - T_{01}) \tag{16}$$

where

Q = heat dissipated to air, Btu/hour-foot - ^{0}F

U = overall heat transfer coefficient, Btu/hour-foot - ^{0}F

A = heat exchanger area, feet

T = temperature, ^{0}F

$_A$ = (subscript) air side of exchanger

$_0$ = (subscript) oil side of exchanger

$_1$ = into exchanger

$_2$ = out of exchanger

[#]Loss in transmission gear only; the reduction gear losses (2.5%) are dissipated in the engine oil cooler.

Maintaining U, TA_2, TA_1, T_{02} and T_{01} constant to increase Q from the Btu equivalent of 2.2 percent of 1375 horsepower to 5.0 percent of 1500 horsepower, the area A must change in direct proportion to the horsepower and efficiency decrement.

$$\frac{(1500 \text{ HP})\ (5\%)}{(1375 \text{ HP})\ (2.2\%)} = \frac{A_{1500}}{A_{UH-1}} \quad \text{or}$$

$$A_{1500} = (A_{UH-1})\ (1.09)\ (2.27) = 2.58\ (A_{UH-1}) \tag{17}$$

It is seen that the heat exchanger weight and the length of the piping will increase in direct proportion to the area (imagine one long pipe as the heat exchanger). The weight would increase in direct proportion to length L ($W_t - \sigma \frac{\pi (D_0 - D_1)^2 L}{4}$) if D_0 and D_1 are held constant. The surface area would also increase in direct proportion to L (if D is held constant, $A = \pi DL$). The pressure drop in the system also increases in direct proportion to L ($\Delta P = 0.080\ f\ L/DV\ Sg$ from page 3, March monthly report.)

Thus the weight will increase in direct proportion to the Btu equivalent of the horsepower efficiency decrement which must be dissipated.

Heat exchanger weight $= W_t = (W_{tUH-1})\ (2.58) = (4 \text{ lb.})(2.58) = 10.03 \text{ lb.}$

where

W_{tUH-1} = weight of UH-1 transmission heat exchanger.

The pressure drop in the heat exchanger will also increase in direct proportion to the horsepower to be dissipated.

$\Delta P_{1500} = (\Delta P_{UH-1})\ (2.58) = (1.2 \text{ psi})\ (2.58) = 3.1$ psi (with standard fluid)

where

ΔP_{1500} = pressure drop of 1500 HP transmission

ΔP_{UH-1} = pressure drop of UH-1 gear transmission

$\Delta P_{ADD} = \Delta P_{Base}\ (1 - \text{F.R. @ } N_{RE})$

N_{RE} = 4350 @ 230°F

F.R. @ 4350 = .54

ΔP_{ADD} = (3.1) (1 - .54) = $\underline{\underline{1.42 \text{ psi}}}$

where

ΔP_{ADD} = pressure drop with additive, psi

ΔP_{Base} = pressure drop of base fluid, psi

F.R. = friction reduction

N_{RE} = Reynolds number

2. Effect of Heat Transfer Coefficient

The reduction of fluid losses in the system reduces turbulence, which in turn reduces the heat transfer coefficient of the oil. A correlation (Equation 24) has recently been experimentally established by The Western Company for friction-reducing additives. It is shown as Figure 4. Using Figure 4 and the friction factor, f, for smooth pipes, a heat transfer coefficient for the treated fluid was calculated:

For N_{RE} = 4350; f - .039 Reference 1 (18)

$f/2 \times 10^{-3} = 19.5$ (19)

at 19.5 $f/2 \times 10^{-3}$; $st\,(P_R)^{2/3} \times 10^3 = 19.5$ (20)

To reduce ΔP by .54, f will reduce by .54 at N_{RE} of additive:

$$N_{RE}\text{ Additive} = N_{RE_{Base}} \frac{(\nu\text{ Base})}{\nu\text{ Additive}} \tag{21}$$

$$N_{RE}\text{ Additive} = 4350\,\frac{(4.25)}{5.42} = 3410 \tag{22}$$

where

ν = kinematic viscosity, pound/foot$_3$ (23)

$f\,@\,N_{RE}$ = 3410 = .042

f @ F.R. = .54 = (.042) (1 - .54)

f = .0/93

$$f/2 \times 10^3 = 8.65$$

$$@\ f/2 \times 10^3 = 8.65; \quad St\,(PR)^{2/3} \times 10^3 = 8.65 \tag{24}$$

$$St\,(PR)^{2/3} \times 10^3 = \frac{h}{C_p \sigma V} \frac{(C_p \mu)^{2/3}}{K} \times 10^3 \tag{25}$$

where

St = Stanton number

PR = Prandtl number

h = heat transfer coefficient, Btu/hour-foot^2 - ^{0}F

C_p = specific heat of oil, Btu/pound - ^{0}F

σ = density, pound/foot^3

μ = viscosity, pound/second-foot

K = conductivity, Btu/hour-foot - ^{0}F

V = velocity, feet/second

To obtain the ratio of the heat transfer coefficient for the base fluid to the fluid with additive, we take the ratios of Equation (25) for both fluids,

$$\frac{\left[\frac{h}{C_p \sigma V}\left(\frac{C_p \mu}{K}\right)^{2/3} \times 10^3\right] \text{Additive}}{\left[\frac{h}{C_p \sigma V}\left(\frac{C_p \mu}{K}\right)^{2/3} \times 10^3\right] \text{Base}} = \frac{8.65}{19.5} \tag{26}$$

holding inlet flow and diameter the same. Since adding the additive to the system does not change certain fluid parameters, this allows cancellation of some of the parameters of Equation (26), and the equation then becomes

$$\frac{(h \sigma \mu^{2/3})\ \text{Additive}}{(h \sigma \mu^{2/3})\ \text{Base}} = .444 \tag{27}$$

substituting $\mu = \nu \sigma$ in (10) (28)

$$\frac{(h\sigma\ (\nu\sigma)^{2/3})\ \text{Additive}}{(h\sigma\ (\nu\sigma)^{2/3})\ \text{Base}} = .444 \tag{29}$$

$$\frac{h_{Add}\ .837\ (.837 \times 5.42)}{h_{Base}\ .830\ (.830 \times 4.25)} = .444 \tag{30}$$

$$h_{Add} = (h)_{Base}\ .444\ (.992)\ \frac{(2.31)}{(2.79)} = .365\ h_{Base} \tag{31}$$

From Reference 3, the overall heat transfer coefficient U has the following relationship:

$$\frac{1}{U_0} = \frac{1}{h_0} = \frac{a}{(A_w/A_h)} + \frac{b}{(A_c/A_0)}\ \eta_A\ h_A \tag{32}$$

where

$a/(A_w/A_h)K$ = wall conductive component

$b/(A_c/A_0)\ \eta_A h_A$ = an air side film connection component including the temperature ineffectiveness of the extended fin area on the air side.

Thus U_0 will be affected less than h_0 is affected by the additives in the fluid, since a, A_w/A_0, K, A_c/A_0, η_A and b will not be changed.

Therefore, to be conservative again, let

$$\frac{U_{Add}}{U_{Base}} = \frac{h_{Add}}{h_{Base}} \tag{33}$$

or

$$U_{Add} = U_{Base}\ \frac{(h_{Add})}{(h_{Base})} \tag{34}$$

Substituting 17 into Equation (1) and solving for A,

$$A_{Add} = \frac{A_{Base}}{.365} \tag{35}$$

Maintaining diameter of exchanger tubes constant,

$$W_{t_{Add}} = \frac{W_{t_{Base}}}{.365} \quad (36)$$

$$W_{t_{Add}} = \frac{10.03}{.365} = \underline{\underline{27.4 \text{ pounds}}} \quad (37)$$

Maintaining a diameter of exchanger tubes constant,

$$\Delta P_{Add} = \frac{\Delta P}{.365} = \frac{1.42 \text{ psi}}{.365} = 3.9 \text{ psi} \quad (38)$$

Thus, by using all conservative approximations and not optimizing the cooler system, we have a heat exchanger which is large but does not have much pressure drop.

If it is elected to optimize the system and to do a more exact sizing, the heat exchanger will probably be between 10 pounds and the 27.4 pounds calculated.

GEAR/SHAFT TRANSMISSION HEAT EXCHANGER DESIGN INFORMATION

1. Transmission Oil Cooler

   a. The heat exchanger is designed for 125°F ambient air temperature. It is tested to MIL-STD-210A hot-day conditions of 103°F temperature. The exchanger is not designed with a surplus heat transfer capacity.

   b. Airflow over the cooler is 90 pounds/minute in hovering mode at 7000 - 9000 feet.

   c. The system is designed to absorb 2.2-percent efficiency decrement of 1375 horsepower at hot-day conditions at 7000 - 9000 feet altitude.

   d. Maximum oil temperature is 230°F.

   e. Heat transfer coefficient of oil in system is 0.52 Btu/hour/foot^2 °F.

   f. Internal oil flow is 8.4 gallons/minute.

   g. Pressure drop in cooler is 1.2 psi.

   h. Heat exchanger size and weight:

Size = 12-1/4" x 2-1/2" x 3.62" deep.
Weight = Maximum of 4 pounds, dry, with fittings.

i. The cooler has:
Five air centers
Eighteen fins 3.75" high x .006"
Four oil centers
Eighth-inch ID tubes/.014" material
Tube effective length = 10.8"

2. Engine/Reduction Gear Oil Cooler

a. Internal oil flow is 12 gallons/minute.

b. Six hundred Btu/minute extracted at hot-day conditions (103°F) at 7000 - 9000 feet altitude (critical altitude) at hover.

c. This oil cooler cools both the engine oil and T-53 reduction gear oil.

Unclassified
Security Classification

DOCUMENT CONTROL DATA - R & D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY (Corporate author) The Western Company of North America, Research Division, 2201 N. Waterview Pkwy., Richardson, Texas 75080	2a. REPORT SECURITY CLASSIFICATION Unclassified 2b. GROUP N/A

3. REPORT TITLE
Investigation of Hydraulic Power Transmission Systems for V/STOL Aircraft

4. DESCRIPTIVE NOTES (Type of report and inclusive dates)
Final Report

5. AUTHOR(S) (First name, middle initial, last name)
Jerome L. Overfield and Horace R. Crawford

6. REPORT DATE	7a. TOTAL NO. OF PAGES	7b. NO. OF REFS
August 1967	84	13

8a. CONTRACT OR GRANT NO. DA44-177-AMC-333(T)	9a. ORIGINATOR'S REPORT NUMBER(S) USAAVLABS Technical Report 67-40
b. PROJECT NO. c. Task 1M125901A01410 d.	9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report) 3702-1

10. DISTRIBUTION STATEMENT
Distribution of this document is unlimited.

11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY
	U. S. Army Aviation Materiel Laboratories Fort Eustis, Virginia

13. ABSTRACT
A comparison is presented between the efficiency and weight of the UH-1 helicopter gear transmission and speed reducer systems to a hydraulic power transmitting system for the UH-1. The hydraulic transmission system is composed of a "turbine speed" hydraulic pump and a hydraulic motor. The hydraulic transmission system utilizes recently developed non-Newtonian, viscoelastic fluids which greatly reduce fluid friction losses, increase hydraulic component volumetric efficiency, and allows a substantial reduction in connecting fluid line sizes without the usual high fluid friction losses which normally result from use of small lines. The results of the comparison show the hydraulic transmission system efficiency closely matches the gear system efficiency, and the hydraulic transmission system is lighter than the gear system. A limited aircraft effectiveness study shows that the flexibility of design which is possible with a hydraulic transmission can result in an increase in useful engine horsepower, reduced infrared signature, improved foreign object damage protection and a reduction in engine/airframe vibration.

The hydraulic transmission system is based upon hydraulic components which were chosen from an industry and government agency wide search for the best hydraulic components.

Design data and calculated and test performance are shown for the hydraulic components. Test data is shown for the fluid flow friction reducing hydraulic fluid.

Vulnerability, maintenance, logistics and operational considerations for a hydraulic transmission are presented. ( )

DD FORM 1 NOV 65 1473 REPLACES DD FORM 1473, 1 JAN 64, WHICH IS OBSOLETE FOR ARMY USE.

Unclassified
Security Classification

Unclassified
Security Classification

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Hydraulic Power Transmissions						
Hydrostatic Transmission						
Transmission						
Non-Newtonian Fluid						
Visco-Elastic Fluid						
Fluid Transmission						
Helicopter Transmission						
V/STOL						
V/STOL Power Transmission						

Unclassified
Security Classification